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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE | READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) Fendering for Structural Steel Dolphins, Naval Facilities Engineering Command (NAVFAC)		5. TYPE OF REPORT & PERIOD COVERED Final of 30 November 1976
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Final rept. Myle J. Holley Jr.		8. CONTRACT OR GRANT NUMBER(s) N00025-74-C-0030
9. PERFORMING ORGANIZATION NAME AND ADDRESS Hansen, Holley & Biggs Cambridge, MS 94 p.		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS Engineering Investigation O&M,N 2571
11. CONTROLLING OFFICE NAME AND ADDRESS 04C Program Coordinator Dept of the Navy, Alexandria, VA		12. REPORT DATE 30 November 1976
		13. NUMBER OF PAGES 94
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office) 04B Consultant, Waterfront Structures, NAVFAC, Dept of the Navy		15. SECURITY CLASS. (of this report) UNCLASSIFIED
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

UNCLASSIFIED AND UNLIMITED

DISTRIBUTION STATEMENT A
Approved for public release;
Distribution Unlimited

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

UNCLASSIFIED AND UNLIMITED

DDC AUG 31 1977

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Energy ranges, Fender forces, Contact areas, Articulations, Rotations, Camels. Timber shield, Tire casings, Tire stack. Mounting of tires, Vessel approach - angle, Hull form.

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

Fendering for Structural Steel Dolphins with various contact areas, horizontal and vertical rotations, fender shields, New concepts of tire casings for surface ships and for submarines. This fendering concepts can also be used outside the Structural Steel Dolphins.

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FENDERING FOR STRUCTURAL STEEL DOLPHINS

NAVAL FACILITIES ENGINEERING COMMAND
DEPARTMENT OF THE NAVY

November 30, 1976

ABSTRACT

Presented is the following Naval Facilities Engineering Command (NAVFAC) publication entitled:

"Fendering for Structural Steel Dolphins".

The contents include:

1. Foreword
2. Preface with introductory remarks
3. The report on fendering with various contact areas, horizontal and vertical rotations, fender shields, new concepts of tire casings for surface ships and for submarines.

This publication was accomplished by the NAVFAC Engineering Investigation (EI) program on 30 November 1976.

FOREWORD

"FENDERIMG FOR STRUCTURAL STEEL DOLPHINS"

The total energy absorbed by the structural system will be the sum of the fendering capability, the structure proper, and the hull of the ship. Usually fenders in ports and harbors consist of low cost and low capacity devices, taxing thus unduly severely the structure itself or the ship. To make the situation still worse, some fenders are geared, primarily, to prevent damage to ship's coating on contact with the herthing structure. Aside from human factors in the past, with timber structures having a high degree of resiliency and with smaller ships, such systems with meager fenderings were generally adequate. However, for other than timber, higher capacity and less resilient structures, serving larger ships with deeper drafts, problems may arise.

Therefore, various concepts regarding "Fendering for Structural Steel Dolphins" are discussed. These concepts could also be considered for use in conjunction with rigid structures.

PREFACE

Introductory remarks to "Fendering for Structural Steel Dolphins"

1. Fender design concepts are presented for steel dolphin structures, having in mind the following ships:

   a. Cargo

   b. Oiler

   c. Ammunition

   d. Cruiser

   e. Destroyer

   f. Submarine

2. The limiting parameters and factors for design of fenders will be water depth, amount of ship's energy to be absorbed, approach contact angle of the vessel, tidal range, and allowable contact unit pressure of fender on ship's hull plates.

3. The design Standards for Structural Steel Dolphins in cohesionless soils dated 20 January 1974 was disseminated to the field previously. It covered single and multiple pile - dolphins 36", 48", 60" and 72" with wall thickness from o.75" for all sizes to 1.5" for 60" and 72" diameter piles, in water depth from 40 to 70 feet.

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LIST OF SYMBOLS

C = Combined coefficient (= $C_E \times C_H \times C_S \times C_c$)

C_c = Configuration coefficient

C_d = Ship deformation coefficient

C_e(= C_E) = Eccentricity coefficient

C_g = Ship geometric coefficient

C_H(= C_m) = Hydrodynamic coefficient

C_M = Pile bending moment coefficient

C_S = Softness coefficient

C_T = Coefficient to account for loaded vs. unloaded vessels

C_x = Coefficient for site exposure

D = Pile diameter, ins., ft.

DWT = Dead weight tonnage, long tons

E(= W) = Vessel kinetic energy to be absorbed, kip-ft.

F = Dolphin force, at rated energy, kips

F/W = Force/energy ratio, kip/kip-ft.

$(F/W)_{struct.}$ = Force/energy ratio for dolphin structure, kip/kip-ft.

$(F/W)_{tires}$ = Force/energy ratio for tire-stack fender, kip/kip-ft.

$(F/W)_{mtg.}$ = Force/energy ratio for fender mounting elements, kip/kip-ft.

f_y = Steel yield stress, ksi.

FLDT = Full load displacement, long tons

M = Vessel mass, kip-sec^2/ft.

t = Pile wall thickness, ins.

V(= V_n) = Vessel velocity component normal to fender, ft/sec.

$W_{struct.}$ = Energy absorbed by dolphin structure, kip-ft.

W_{tires} = Energy absorbed by tire-stack fender, kip-ft.

$W_{mtg.}$ = Energy absorbed by rubber mounting element, kip-ft.

FENDERING FOR STRUCTURAL STEEL DOLPHINS

1. INTRODUCTION

Contract N00025-71-0023 Design Standards for Structural Steel Dolphins has developed steel dolphin structures suitable for a range of ship sizes and velocities, in water depths from 40 to 70 feet. It also developed methods and design aids for the design of steel dolphin structures for other mooring force requirements, energy requirements, and for any specified steel yield strengths. However, contract N00025-71-0023 focused upon mooring strength and elastic energy capacity of the steel dolphin structure, without consideration of fendering.

The present contract is concerned with fendering for structural steel dolphins of the kind developed in contract N00025-71-0023. These structures are comprised of cylindrical steel piles cantilevered from the seabed. Where the dolphin is comprised of multiple piles, the intra-pile connections, at the top of the dolphin, constrain the several piles to undergo essentially equal deflections, even when the applied loading is eccentric to the dolphin axis. This feature insures equal distribution of load, and elastic energy, among the piles; i.e., efficient utilization of the capacities of all the piles, even under eccentric loading.

The design of dolphin fendering involves the following questions:

a) For the range of vessel sizes and approach velocities to be served, and for conditions at the specific facility served (water currents, waves, winds, water depths and tidal range, vessel approach angles), what is the appropriate number and location of dolphins, and what is the energy to be absorbed by each dolphin?

b) At the acceptable levels of fender-to-hull pressures, and reasonable contact areas, what are the maximum acceptable fender forces for the vessels to be served?

c) Is it feasible for each of the required dolphins to absorb all of its assigned energy solely by elastic strain of the steel piles, and at maximum fender force within the hull force limits?

d) If additional energy absorption (beyond the elastic strain energy capacity of the piles) is required, either to satisfy the energy demand per se or to satisfy the energy demand at acceptable force levels, what energy-absorbing devices are appropriate?

e) If energy-absorbing devices for the excess must be added, where are they best located; i.e., at the vessel-to-fender interface, or within the fender mounting assembly?

f) What articulations need be incorporated in fender mountings to accommodate tidal ranges, vessel approach angles, variations in hull geometry, and tilting of the dolphin axis under load?

g) What fender mounting design details best provide the necessary articulations (f)?

It will be apparent that some of the above questions, particularly question (a), go beyond the matter of fender development, and encompass the entire dolphin design problem. The choice of dolphin numbers, locations and sizes will vary from site to site, and with the operational requirements of the different facilities served. Accordingly, in the present contract it is necessary to assume a range of possible conditions, rather than conditions specific to hypothetical sites.

For the six classes of vessels required to be considered, information has been obtained on displacement tonnages and on limiting fender-to-hull pressures. Based upon these data, it has been possible to determine the range of energies to be absorbed and the fender force limits imposed by hull pressures and reasonable fender contact zone geometries. The principal focus of the effort covered by this report has been upon the development of fenders and fender-to-dolphin connections suitable for the range of conditions of interest.

For most of the vessel classes of interest the limitations on hull pressures lead to requirements for large contact areas. This fact, together with the large approach angles (20° for destroyers and smaller vessels; 10° for larger vessels), specified on page 26-5-4 of NAVFAC DM-26, require the

use of a fender shield in most cases. An effort has been made to develop a means of mounting a conventional timber shield that will incorporate the necessary articulations, and also incorporate additional energy-absorbing elements when these are required.

Because of the severe vertical curvature of nuclear submarine hulls, a rigid timber shield is not a particularly appropriate fender choice. Here, and for other classes of vessels berthing under moderate conditions of approach angle and velocity, a fendering system using tire casings seems to be attractive. Our recommendation was strongly influenced by the successful experience of the Toronto Harbour Commission with simple rotating fenders incorporating used, large diameter, tire casings packed with shredded rubber. These fenders are simple, inexpensive, and (because they are free to rotate) they reduce the vessel-to-fender friction force to negligible magnitude. The Toronto concept is modified to accommodate the more corrosive ocean environment, as well as its mounting, to accommodate additional energy-absorbing elements.

2. SCOPE

This contract requires the development of fender designs suitable for dolphins servicing the following classes of vessels:

cargo ship	cruiser
oiler	destroyer
ammo ship	submarine*

under a wide range of site conditions (water depth, currents, waves, wind) and vessel operating conditions (approach angles and velocities). In order to reduce these many variables to practical design parameters, the following controlling parameters are identified:

a) Water depth
b) Energy to be absorbed
c) Contact angles
d) Tidal range
e) Hull limits to fender force.

Water depth is a critical parameter because the elastic energy capacity of a steel dolphin of any given configuration increases (and the associated maximum fender force decreases) with increasing water depth.

Contact angles (reflecting hull form at the point of fender contact, the vessel approach angle and dolphin slope changes) and tidal range influence the required degree of articulation in the fender-to-dolphin connection.

Hull limits to fender force, in combination with required energies, determine whether energy-absorbing devices must be added. Limits to hull pressure also influence the required size of the fender shield (or, more generally, the area of the vessel-to-fender contact zone).

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* The Preliminary Report distinguished between diesel and nuclear submarines. In this Final Report the former category has been omitted, since we are informed that few diesel submarines remain.

This report presents:

1. A discussion of factors influencing dolphin fender design.
2. Recommended dolphin fender design concepts suitable for the six vessel classes over a range of the above five parameters.
3. Interrelationship of fender choices and major steps in the dolphin design process.
4. Recommendations for certain development efforts.

3. VESSEL CLASSES AND ASSOCIATED ENERGY RANGES

The Navy provided information on full load displacement tonnages which is presented in Table 1, below.

Table 1 - Full-Load Displacement Tonnages

Vessel Class	FLDT (Long Tons	
Cargo	20,000	
Cargo (Combat Stores)	53,000	(AOE-1 53.6 LT @ 39')
Oiler	40,000	
Ammo	20,000	
Cruiser (Heavy)	21,000	
Destroyer	5,200	
Submarine (Nuclear)	7,000	(SSBN-616 8.25 LT @ 33')

The Navy suggested approach velocity components, normal to the fender, of 0.5 ft/sec and 1.0 ft/sec, respectively, for sheltered and unsheltered sites. These may be compared with the values given on Figs. 2-22, 2-23, and 2-24 of NAVFAC DM-25. The latter specify, for the range of site exposures, 0.2 to 0.8 ft/sec. In this study we have adopted values of 0.3, 0.6 and 0.9 ft/sec for lower bound, average, and upper bound conditions. Since the energy demand is proportionate to the velocity squared, the demand for any other velocity easily can be computed.

The Navy suggested the use of a single modifying coefficient to account for the flexibility of the "deck." Such a coefficient is not unreasonable in the design of energy-absorbing fender devices for use on pier structures. In the case of dolphins, however, there is no "deck" structure to absorb energy.

The energy to be absorbed by a dolphin, together with energy-absorbing fender devices if these are added, is a function of several variables. These include, but are not limited to, the vessel mass, velocity, point of impact on the hull, effective water mass moving with the hull, and elastic energy absorbed by hull deformation. Ref. (2) suggests the following form for the energy equation, accounting for the principal variables, when only translational motion is considered.

$$W = 0.5\ MV_n^2 \times C_E \times C_H \times C_S \times C_c \qquad (1)$$

where: W = Kinetic Energy, kip-ft.

M = Vessel, Mass, kip-sec^2/ft.

V_n = Translational Velocity Normal to Fender, ft./sec.

C_E = Eccentricity Coefficient

C_H = Hydrodynamic Coefficient

C_S = Softness Coefficient

C_c = Configuration Coefficient.

The eccentricity coefficient, C_E, is a function of the radius of gyration of the ship about a vertical axis through the center of gravity, distance from the center of gravity to the point of impact, and angle between the velocity vector and the radial line from center of gravity to point of impact. Values of C_E as large as 1.0 are possible, but the value 0.5 is recommended as most representative.

Based on a study of 70 U.S. Naval vessels in the 2000 - 20,000 long-tons class, Lee, Ref (3), suggests C_H values of 1.15 to 1.7. These appear consistent with the values 1.74 to 1.84 suggested in Ref (2) for vessels in the 50,000 - 500,000 ton class.

Since, as already noted, the dolphin and any attached energy-absorbing devices must absorb essentially all the energy (i.e., there is no pier to absorb part of the energy) it appears prudent to adopt a value C_S = 1.0 for the softness coefficient.

Values of the configuration coefficient, C_c, (Refs 2 and 3) are 1.0 for an open pier, 0.9 for a semi-closed pier, and 0.8 for a closed pier. Since these are applicable to fendering mounted directly against piers, and dolphins do not present a "closed" configuration, the value C_c = 1.0 appears to be appropriate.

When the above values of individual energy correction coefficients are combined, an overall correction coefficient, C, results:

$$C = C_E \times C_H \times C_S \times C_c \quad (2)$$

$$= 0.5 \times 1.67 \times 1.0 \times 1.0 = \underline{C = 0.83} \quad (3)$$

By substituting (3) into (1), we obtain

$$W = 0.5\ M\ V_n^2 \times C$$

$$= 0.42\ M\ V_n^2 \quad (4)$$

NAVFAC DM-25 (pp. 25-2-33 through 25-2-36) expresses the effective berthing energy, E, of a ship as

$$E = C[(C_m\ M)\ \frac{V^2}{2}] \quad (5)$$

where $C = C_e\ C_g\ C_d\ C_c$ (6)

and E = effective berthing energy, kip-ft.

M = ship mass, kip-sec^2/ft.

V = normal component of approach velocity, ft/sec.

C_m = a coefficient to account for the mass of water moving with the ship

C_e = eccentricity coefficient

C_g = ship geometric coefficient

C_d = ship deformation coefficient

C_c = berth configuration coefficient.

The above defined coefficients C_m, C_e, and C_c are exact equivalents of the coefficients C_H, C_E, and C_c appearing in Eq. (1). The above coefficient, C_g, has no equivalent in Eq. (1), but the NAVFAC DM-25 development recommends a conservative value, C_g = 1.0. The above defined coefficient, C_d, accounts for energy absorbed by deformation of the ship hull, whereas C_S in Eq. (1) accounts for energy absorbed in the ship hull and in the pier structure. NAVFAC DM-25 recommends a value C_d = 1.0 and, as noted, we have taken C_S = 1.0. From this comparison it will be apparent that the NAVFAC DM-25 procedure leads to values of effective berthing energy which are essentially identical with that expressed by Eq. (4). The NAVFAC energy term E corresponds to the energy term W in Eq. (1).

It will be apparent that the assumed velocity, V_n, is of dominant importance. Based on 650 measured tanker impacts at three different facilities, the following alternative equation for energies with a low probability of exceedance is recommended in Ref. (2).

$$W = .0168 \times DWT \times C_T \times C_x \qquad (7)$$

where: DWT = Dead Weight of Vessel (Long Tons)

C_T = 0.85 for a Loading Terminal
= 1.00 for an Unloading Terminal

C_x = 1.00 for a Sheltered Harbor
= 1.18 for a Normally exposed Harbor
= 1.30 for a Very Exposed Harbor.

Converting DWT to mass, M, Eq. (7) becomes

$$W = 0.24\ M \times C_T \times C_x \qquad (8)$$

Equating (4) and (8), we obtain values of velocity, V_n, which, when inserted in Eq. (4), would yield the same energies as the statistically based Eq. (7); i.e.,

$$V_n = \sqrt{\frac{0.24\ C_T\ C_x}{0.42}} \qquad (9)$$

Eq. (9) yields the following values of V_n corresponding to the recommended values of C_T and C_x

C_T	C_x	V_n
0.85	1.00	0.70 ft/sec
0.85	1.18	0.76 ft/sec
0.85	1.30	0.80 ft/sec
1.00	1.00	0.76 ft/sec
1.00	1.18	0.82 ft/sec
1.00	1.30	0.87 ft/sec

On the basis of comparisons of the above kind, Ref. (2) suggests that the common assumption that V_n = 0.5 ft/sec may be unconservative in many cases. On the other hand, it must be remembered that the statistically

based Eq. (7) probably reflects impact measurements on tankers most of which were much larger vessels than those which are of interest to the present studies. More important, the above comparison seems to provide some degree of confirmation of the validity of Eq. (4) for upper-bound energy estimates based on V_n = 0.9 ft/sec. The comparison also seems to suggest that while an assumed V_n = 0.3 ft/sec may provide a reasonable lower bound, average energy values may be more prudently based on an assumed V_n = 0.6 ft/sec.

Based on Eq. (4), the vessel tonnages of Table 1 and "low," "average," and "high" velocities of 0.3, 0.6 and 0.9 ft/sec, vessel energies are presented in Table 2. It must be emphasized that dolphin design for any specific facility will require the evaluation of vessel energies for the specific site, vessels to be served, and operating conditions. Such an evaluation may lead to lower, or to higher energy values than those presented in Table 2. It would be highly uneconomical to standardize dolphins and fendering to envelop the wide range of conditions that may be encountered. Thus the purpose of the energy values in Table 2 is not to serve as the basis of a single fender system applicable to all conditions. Rather, these tabulated values serve to indicate the range of energy demands for which dolphins and associated fendering may be required. They are a useful guide to kinds of fendering which may be applicable.

Table 2 - Range of Possible Energy Demands on Dolphins and Associated Fendering*

	ENERGY DEMANDS (ft-kips)		
	LOWER BOUND	AVERAGE	UPPER BOUND
VESSEL CLASS	V_n = 0.3 ft/sec.	V_n = 0.6 ft/sec	V_n = 0.9 ft/sec
Cargo	58	232	524
Cargo (Combat Stores)	154	616	1390
Oiler	116	464	1048
Ammo	58	232	524
Cruiser (Heavy)	61	244	550
Destroyer	15	60	135
Submarine (Nuclear)	20	81	182

*based on Eq (4).

It will be apparent that the tabulated upper bound energy values, for the heavier vessels, cannot be satisfied by the elastic energy developed in the piles of a single standard steel dolphin of the sizes and types presented in Ref. (1). Indeed, single standard steel dolphins of the types given in Ref. (1) would not satisfy the average energy values tabulated for the two heaviest vessels. However, the following points should be noted.

1) In many cases, an evaluation of the specific site may lead to lower values than those tabulated, particularly when tug assistance to the docking vessel and/or avoidance of severe weather conditions is contemplated.

2) The use of steels of higher grade than the 60 ksi yield steels used in the standard dolphins of Ref. (1) may be economically attractive. Rated energies are nearly proportionate to the square of the pile stress level. See example in Appendix A.

3) When necessary, special camels may be appropriate for the purpose of distributing the vessel energy to two or more dolphins.

4) When necessary, the rated energy capacities of standard steel dolphins Ref. (1) may be augmented by the addition of energy-absorbing devices, either at the fender-to-vessel interface or incorporated within the fender mounting.

4. REQUIRED FENDER FORCES AND FENDER CONTACT AREAS

4.1 Dolphin Force/Energy Ratios

The fender force required to develop the rated energy of a steel dolphin is a function of the number, diameter, and thickness of the piles, the pile stress level at rated energy, the characteristics of the seabed soil, and the water depth. Since all piles in a dolphin normally are of the same diameter and thickness, the ratio of force to energy will be essentially the same for a single pile as for the dolphin pile group. Thus the force/energy ratio can be examined in terms of the behavior of a single pile.

In general the force/energy ratio increases (but not rapidly) with increasing pile diameter and decreases with increasing pile wall thickness. The ratio decreases in inverse proportion to the design level of pile bending stress. This decrease reflects the fact that the absorbed energy is proportionate to the square of the stress level, while the force is proportionate to the first power of the stress level. The standard steel dolphins recommended in Ref. (1) are based upon a 60 ksi yield steel. However, some overseas dolphin manufacturers use steels of up to 100 ksi yield, and Ref. (4) emphasizes the potential advantages of such steels. The fact that the energy rating increases with the square of the stress level (leading to a smaller number of piles per dolphin, and simplified inter-pile connections) may more than offset the higher pound price of higher grade steels. In the context of the present study two points should be noted. First, the use of higher grade steels can greatly increase the range of vessel energies (i.e., vessel tonnages and velocities) which can be accommodated by steel dolphins. Thus such steels probably should be considered at installations where the vessel energies to be absorbed are particularly demanding. Second, the lower force/energy ratios associated with higher pile stress levels reduces the likelihood that special energy-absorbing devices will be needed, and reduces the energy capacities that such devices must provide if they are needed.

Ref. (1) provides design aids which facilitate the rapid determination of the force/energy ratio for a pile of any given diameter and thickness,

any yield stress grade, and in any water depth. It may be noted that (since a linear force-displacement ratio is conservatively assumed for the pile), the force/energy ratio is simply twice the inverse of the pile deflection at rated energy. In using the design aids of Ref. (1) to compute this ratio, one assumes the upper value of the range of possible soil stiffnesses for the particular site. See example in Appendix A.

Table 3 presents typical force/energy ratios for the two pile diameters, (36" and 48"), used in the recommended standard steel dolphins, two pile thicknesses, (0.75", as recommended for the standard dolphins, and 1.5"), and for the 60 ksi yield steel used in the standard dolphins. Table 4 presents force/energy ratios for the same pile diameters and thicknesses, but for a 100 ksi yield steel.

It should be noted that the dolphin force/energy ratios in Table 3 compare favorably with the corresponding ratios for commercially available energy-absorbing fender devices, particularly at the greater water depths. Obviously the ratios listed in Table 4 compare even more favorably with the corresponding ratios of commercial devices. However, it must be recognized that the performance of an energy-absorbing <u>fender device</u> is independent of the water depth. In contrast, both the rated energy and the force/energy ratio of a steel dolphin are <u>very</u> sensitive to water depth; i.e., the rated energy increases and the force/energy ratio decreases with increasing water depth. For these reasons the use of special energy-absorbing devices is more apt to be necessary in shallow water installations. Fortunately, facilities serving the largest vessels may not be sited in very shallow water.

Table 5 shows fender forces corresponding to the "average" vessel energies of Table 2, and the force/energy ratios of Table 3 for 0.75" pile thickness and for 36" and 48" pile diameters. From an inspection of Table 2 it is apparent that the corresponding fender forces for the lower-bound and upper-bound vessel energies are simply 25 percent and 225 percent, respectively, of the force values shown in Table 5. Table 6 is similar to Table 5, but is based on the force/energy ratios in Table 4; that is, it assumes that 100 ksi yield grade steel is used for the piles.

Table 3 - Force/Energy Ratios at Dolphin Rated Energies
(60 ksi yield steel)

WATER DEPTH	FORCE/ENERGY RATIOS (kips/kip-ft) D = 36" t = .75"	D = 36" t = 1.5"	D = 48" t = .75"	D = 48" t = 1.5"
40'	1.74	1.41	2.02	1.59
50'	1.31	1.08	1.55	1.25
60'	1.02	0.85	1.22	1.00
70'	0.82	0.68	0.99	0.82
80'	0.67	0.56	0.81	0.68

Note: See example computation in Appendix A.

Table 4 - Force/Energy Ratios at Dolphin Rated Energies
(100 ksi yield steel)

WATER DEPTH	FORCE/ENERGY RATIOS (kips/kip-ft) D = 36" t = .75"	D = 36" t = 1.5"	D = 48" t = .75"	D = 48" t = 1.5"
40'	1.04	0.85	1.21	0.95
50'	0.79	0.65	0.93	0.75
60'	0.61	0.51	0.73	0.60
70'	0.49	0.41	0.60	0.49
80'	0.40	0.34	0.49	0.41

Note: See example computation in Appendix A.

Table 5 - Required Fender Forces Corresponding to Vessel Energies[1,2,3]

(f_y = 60 ksi)

VESSEL CLASS	Depth / Pile Dia.	FENDER FORCES (kips) H = 40' 36" 48"	H = 50' 36" 48"	H = 60' 36" 48"	H = 70' 36" 48"	H = 80' 36" 48"
Cargo		404- 469	304-360	237-283	190-230	155-188
Cargo (Combat Stores)		1070-1353	806-953	627-750	504-609	412-498
Oiler		807- 937	608-719	473-566	380-459	311-376
Ammo		404- 469	304-360	237-283	190-230	155-188
Cruiser (Heavy)		425- 493	320-378	249-298	200-242	163-198
Destroyer		104- 121	79- 93	61- 73	49- 59	40- 49
Submarine (Nuclear)		141- 164	106-126	83- 99	66- 80	54- 66

Note (1) Based on Pile t = .75"

Note (2) Based on Average Vessel Energies; multiply by 0.25 or by 2.25 for Lower-Bound or Upper-Bound Energies, respectively.

Note (3) 60 ksi Yield Grade Steel

Table 6 - Required Fender Forces Corresponding to Vessel Energies[1,2,3]

(f_y = 100 ksi)

VESSEL CLASS	Depth / Pile Dia.	FENDER FORCES (kips) H = 40' 36" 48"	H = 50' 36" 48"	H = 60' 36" 48"	H = 70' 36" 48"	H = 80' 36" 48"
Cargo		242-281	182-216	142-170	114-138	93-113
Cargo (Combat Stores)		642-812	484-572	376-450	302-365	247-299
Oiler		484-562	365-431	284-340	228-275	187-226
Ammo		242-281	182-216	142-170	114-138	93-113
Cruiser (Heavy)		255-296	192-227	149-170	120-145	98-119
Destroyer		62- 73	47- 56	37- 44	29- 35	24- 29
Submarine (Nuclear)		85- 98	64- 76	40- 59	40- 48	32- 40

Note (1) Based on Pile t = .75"

Note (2) Based on Average Vessel Energies; multiply by 0,25 or by 2.25 for Lower-Bound or Upper-Bound Energies, respectively.

Note (3) 100 ksi Yield Grade Steel

4.2 Special Measures to Augment Dolphin Energy, and Reduce Fender Force

Comparison of the vessel energies of Table 2 with the rated energies of the standard steel dolphins proposed in Ref. (1) discloses that it may be difficult to service the largest vessels with the standard dolphins, (particularly in shallow waters), unless energy-absorbing devices are added. However, this judgement must be tempered by the following considerations:

1) The "average" estimates of vessel energies may be far too high for conditions at many facilities of interest. Lower-bound energies may be more representative of typical conditions.

2) As has been explained, the use of higher yield grade pile steels may prove an economical method to achieve very much larger dolphin rated energies.

3) The rated energies of the standard dolphins are conservatively based on a maximum pile bending stress of 0.75 f_y (and a pile wall thickness reduced for corrosion losses). It may be sound engineering to allow the bending stresses to reach 1.0 f_y under relatively rare combinations of heavy vessels and adverse operating conditions. It should be noted that the energy-absorbing capacity is increased by more than 67 percent as the bending stress increases from 0.75 f_y to 1.0 f_y.

4) There is a wide variety of energy-absorbing devices which can be added, if necessary, to augment the energy-absorbing capacity of the dolphin piles.

One or more of the above measures will serve to provide the necessary energy absorption, even in the most demanding circumstances. Here it is important to note that many factors which increase energy capacity provide the added advantage of decreased fender force. Thus increased water depth is associated with greater energy capacities and, (as is apparent in Tables 5 and 6), also is associated with lower fender forces. Increased steel yield grade is associated with a large increase in energy capacity; a comparison of Tables 5 and 6 shows clearly that an increase

in steel grade also leads to substantially lower fender forces. Obviously any factor which leads to greater displacements at rated energy implies smaller fender forces. Thus any energy-absorbing device which is added, at the fender-to-hull interface or within the fender mounting, must reduce the fender force. For this reason there may be circumstances when it is desirable to design a dolphin with energy-absorbing device(s) to achieve a lower fender force, rather than to increase the energy capacity of the dolphin.

4.3 Required Fender Contact Areas

The Navy provided information on acceptable fender-to-hull contact pressures for the vessel classes of interest. This data is presented in Table 7. When the fender incorporates a shield, an objective of the mounting should be to assure that the contact pressures are distributed over the shield area; that is, to avoid severe edge or corner pressures. This results in minimum required shield area, and permits the use of simple timber shields. Such shields can be fabricated from 12"-square timbers to areas of at least 150 sq. ft. Table 7 shows acceptable total hull forces, corresponding to the acceptable hull pressures, for contact areas of 50, 100, and 150 sq. ft. These areas can be provided by flat shields, for all classes of vessel except the submarines. The curved hull form of the latter precludes effective pressure distribution over a large flat shield. It is important to note that NAVFAC DM-25, on page 25-2-38, gives acceptable hull pressures for vessels from 15-20,000 tons, which are far in excess of the values recommended by the Navy. Values of acceptable total hull forces, based on the NAVFAC DM-25 allowable hull pressures, are shown in parentheses in Table 7.

An important conclusion can be drawn from a comparison of the acceptable forces in Table 7 with the required forces shown in Tables 5 and 6, supplemented by the discussion in Section 4.2. <u>For dolphins designed to provide the required energy absorption (incorporating energy-absorbing devices if needed), the maximum fender forces do not require exceptionally large contact areas.</u> In most cases a simple timber shield will provide the required area, but fenders which obviate the shield also merit consideration.

Table 7 - Acceptable Hull Contact Pressures and Corresponding Fender Forces

VESSEL CLASS	ACCEPTABLE CONTACT PRESSURE	ACCEPTABLE FENDER FORCE (kips) 50 sq. ft.*	100 sq. ft.*	150 sq. ft.*
Cargo	20 psi	144 (252-360)	288 (504-730)	432 (756-1080)
Cargo (Comtat Stores)	24 psi	173 (" - ")	346 (" - ")	518 (" - ")
Oiler	24 psi	173 (" - ")	346 (" - ")	518 (" - ")
Ammo	20 psi	144 (" - ")	288 (" - ")	432 (" - ")
Cruiser (Heavy)	20 psi	144 (" - ")	288 (" - ")	432 (" - ")
Destroyer	12 psi	86	173	259

* Contact Areas

Note: Values in ( ) are based on allowable hull pressures given in NAVFAC DM-25, p. 25-2-38.

5. ARTICULATION REQUIREMENTS FOR FENDERS ON STEEL DOLPHINS

There are some applications involving small tidal ranges, protected sites, and very narrow ranges of vessel approach angles. For example, the vessel may approach virtually parallel to the face of a dolphin-protected pier, and under tug assistance. Under such conditions articulation may be necessary only to account for the change in slope of the dolphin axis as it deflects. More generally, however, it will be necessary to accommodate variations in the relative position and orientation of the vessel-to-fender contact zone with respect to the dolphin structure. If the fender contact element is an essentially rigid shield, articulations will be provided in the shield-to-dolphin mounting elements. If the contact element is flexible (e.g., tire casings) some of the accommodation may be provided within the contact element itself.

5.1 Vertical Displacements Associated with Tides

If the tidal range is small, and the vessels served can accept fender contact over this range, the fender should be mounted at a fixed elevation. If the tidal range is large, it is possible to use two fenders, mounted at two different fixed elevations, but this may present certain difficulties. Depending on the hull form, and the magnitude of the relative dolphin displacements at the two elevations, both fenders may be contacted, and the distribution of load between the two fenders may be highly indeterminate. Moreover, under high tide conditions the lower fender may be wholly submerged, and its condition not apparent during contact. This does not seem to be a desirable operating condition. Where the tidal range is large, it appears desirable to use a fender which is attached to the dolphin, but which rises and falls with the tide, contacting the vessel hull near the waterline. The mounting must prevent excessive pitching of the fender, under wave action, and must, of course, provide the necessary angular articulations.

5.2 Horizontal Rotations Associated with Vessel Approach Angle, Hull Form, and Dolphin Twist

Under some conditions the range of vessel approach angles is very small, and contact occurs within the vessel quarter points. In such cases, only very little, or no horizontal angular displacement of the fender contact zone relative to the dolphin may occur. NAVFAC DM-26, on p. 26-5-4, indicates a 20° angle of approach for destroyers and smaller craft; 10° for larger vessels. Moreover, initial contact may occur foreward of the foreward quarter point, where the hull is curved; during contact the vessel may be expected to slide along the contact zone, and the latter must accommodate to the hull form.

It should be noted that steel dolphins are, themselves, highly resistant to moments about their vertical axes. Thus the dolphin displaces readily, but will rotate only very little in the horizontal plane, even under eccentric loading. This characteristic is inherent in a single-tube dolphin, since the torsional stiffness is far in excess of the (cantilever) bending stiffness. Multiple-pile dolphins are deliberately designed to display this same torsional stiffness, in order to fully exploit the (flexural) energy-absorbing capacity of all piles. Since the dolphin structure cannot accommodate the range of horizontal angles associated with the vessel approach angles and varying hull form, the accommodation must be provided by articulation in the fender mounting and/or by flexibility of the contact element itself.

5.3 Vertical Rotations Associated with Vessel Roll, Hull Form, and Dolphin Bending

For most vessels, particularly if contact with the fender shield occurs only near the water line, variation in vertical slope of the hull is relatively small. Similarly, for most operating conditions, variations in vertical slope caused by vessel roll will be relatively small. On the other hand, bending of the dolphin piles, as the dolphin displaces, produces a significant change in slope of the dolphin axis at the elevation of the fender mounting. The contact element and/or the mounting must accommodate

this varying inclination of the dolphin axis, plus the (typically smaller) slope changes associated with vessel roll and with differing hull forms.

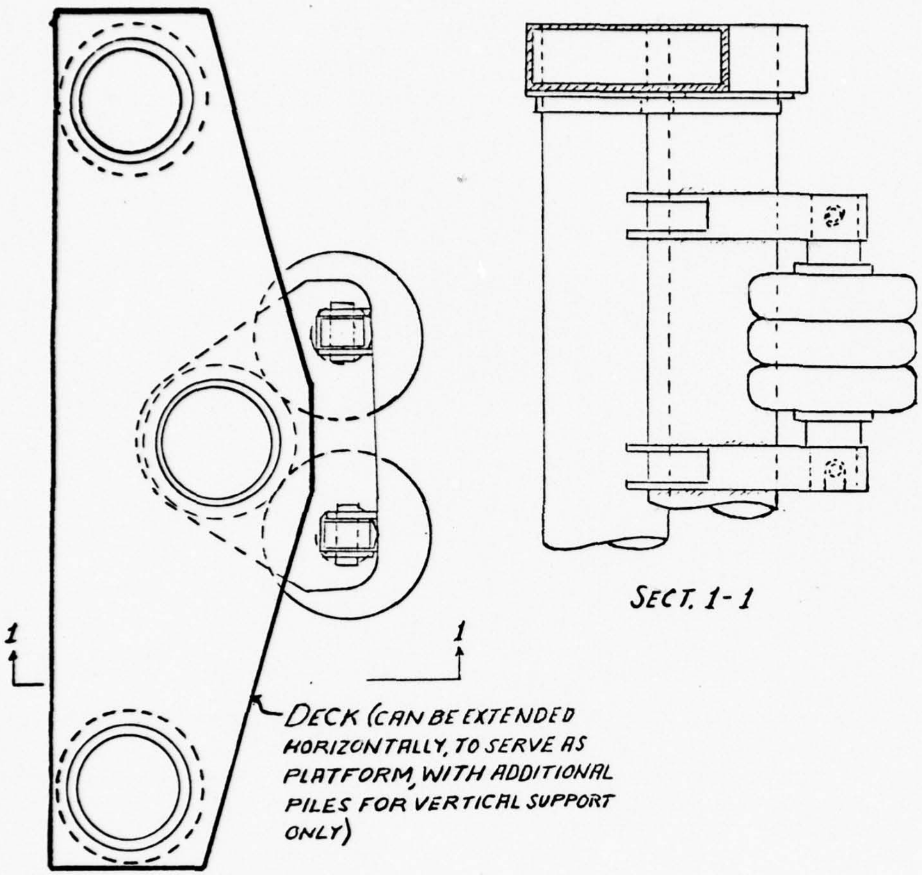
It will be understood that fender mounting details providing for one, or two, of the above-discussed, relative shield-to-dolphin motions, are more readily designed than details incorporating all three components of articulation. In particular, if the tidal range is small enough to permit mounting the fender at a single, fixed, elevation on the dolphin, this may simplify the connection detail.

When a rigid fender shield is provided, its purpose can be defeated if contact occurs only at an edge (or corner) of the shield. It is to avoid just such severe line (or point) contact loads that angular articulation must be incorporated in the fender mounting.

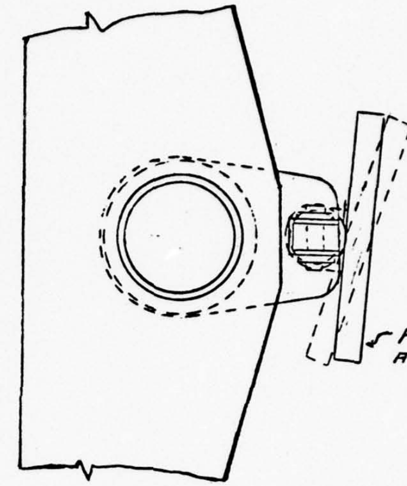
5.4 Location of Fender Mountings Relative to Dolphin Perimeter

The fender contact zone must be far enough from the dolphin to preclude any possibility of contact between any portion of the vessel hull and the submerged portion of the dolphin piles(s). Such contact could be severely damaging to vessel and/or dolphin.

For single pile dolphins and for multi-pile dolphins having piles connected by torque arms, there is essentially no restriction on the location of fender mountings around the perimeter. However, unless there is a sound reason to do otherwise, it is preferable to so locate the mountings as to minimize the eccentricity of loading with respect to the dolphin axis. Under some conditions (particularly when there is a restricted range of vessel approach angles), it is possible to design a multi-pile dolphin without connecting the piles through torque arms. In these dolphin configurations (see Fig. 1), the fender mounting must be located directly on a central pile, which is in a symmetry plane of the dolphin and close to the centroid of the group of piles. Any eccentricity moment associated with the applied force is resisted directly, and entirely, by the central pile. A distribution beam (or truss, or deck) engages the piles, through loose-fitting collars, thereby forcing equal deflections, and, thus, equal sharing of the loading among all the piles.



SECT. 1-1



NOTE

IN THIS TYPE OF DOLPHIN VIRTUALLY ALL OF THE TORQUE IS RESISTED BY THE CENTRAL PILE. THE TIRE-STACK FENDER DEVELOPS MUCH LESS TORQUE, AND THUS IS PREFERABLE TO THE SHIELD FENDER.

HANSEN HOLLEY AND BIGGS		
SCALE:	APPROVED BY:	DRAWN BY
DATE 12-6-76		REVISED
FIG. 1- FENDER LOCATION ON MULTI-PILE DOLPHINS WITH PILES CONSTRAINED IN DECK COLLARS		
		DRAWING NUMBER FIG 1

4.4 Camel-Loaded Dolphins

Under certain restricted conditions (sheltered site, narrowly limited approach angles), it may be attractive to load multiple dolphins through camels or pontoons. In such cases no fender shields are required, and the question of articulation does not arise. All that is required is a suitable bearing surface (plate, or beam) mounted on the outboard side of the dolphin, and a means to restrict longitudinal displacements of the camel and restraints to prevent separation of the camel from dolphins. (See Fig. 2).

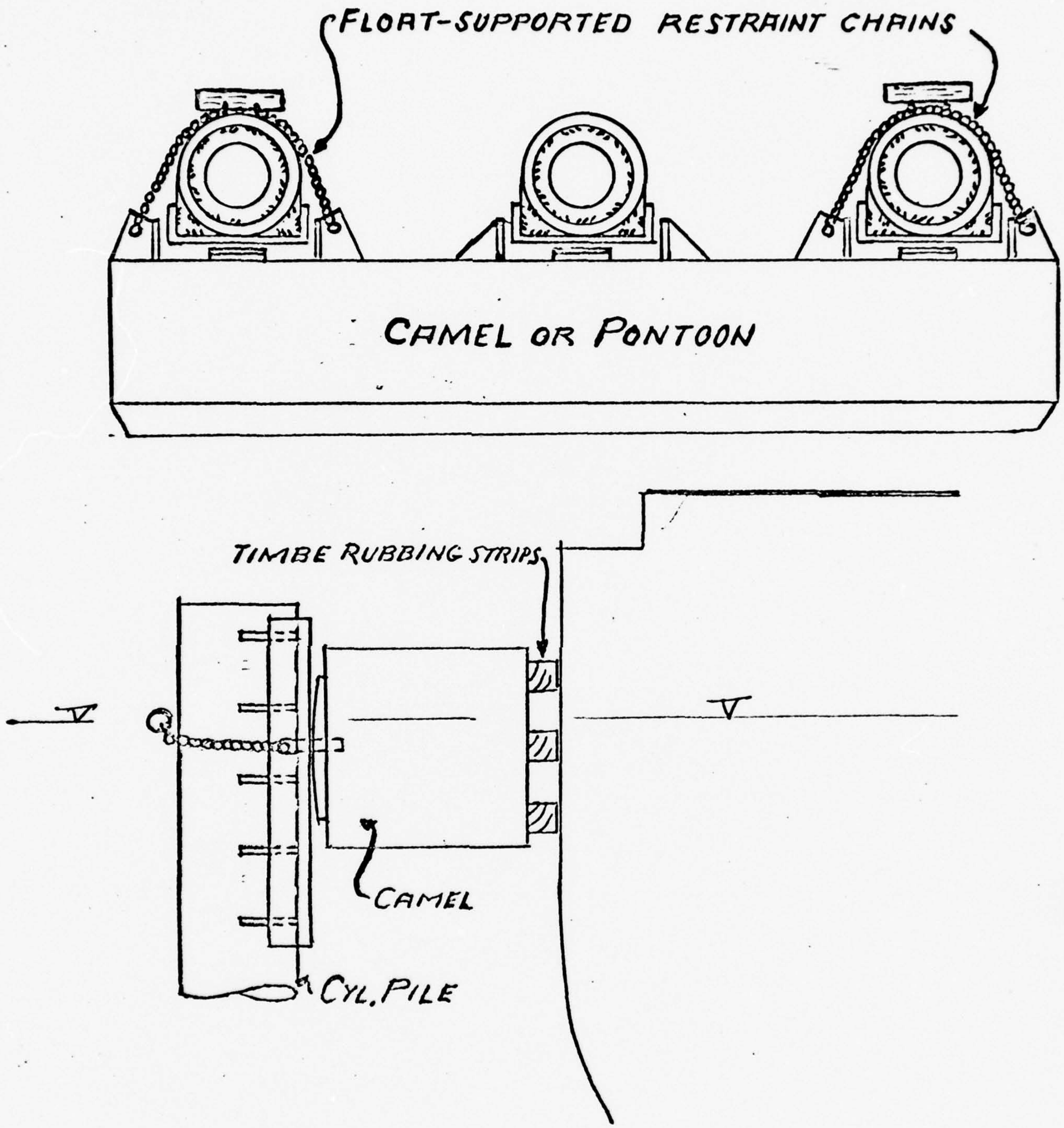


FIG. 2- CAMEL- LOADED DOLPHINS

FIG. 2

6. TIMBER SHIELD DOLPHIN FENDERS

6.1 Objectives

Timber shields have long been used for fendering. They have the advantages of simplicity and easy repair and replacement. In most previous applications the mountings have not been provided with any deliberate angular articulation. There are exceptions, however, in which one degree, or two degrees, of articulation were provided. We have not found cases where both angular articulation and adaptability to tidal variations were incorporated in the shield mounting. Moreover, where angular articulation was provided the precision of the connections used for this purpose appeared often to be excessive and/or unnecessarily demanding, from the point of view of maintenance.

It has been our objective to develop shield mountings which will accommodate all linear and angular shield-to-dolphin relative motions. It has been our further objective that the mountings be simple, economical, not dependent upon close mechanical tolerances, and not unduly sensitive to the corrosive environment within the ocean water or in the splash zone. Finally, it has been our objective to develop mounting details in which energy-absorbing elements can be incorporated when the additional energy absorption thus provided is necessary and/or advantageous.

6.2 Description of the Fender Shield and Details of Mountings

Figs. (3), (4), and (5) show proposed fender shield and mountings. The shield is a simple flat, rectangular, timber element comprised of pine timbers (typically 12" x 12"), assembled and connected to the necessary hardware by recessed bolts. The area of the shield is simply the largest contact force associated with energy absorptions required by the range of vessels to be served, divided by the corresponding permissible hull contact pressures. As noted in Chapter 4, shields of practical sizes (less than 150 ft^2) will be sufficient for much of the range of interest, even without energy-absorbing elements in the fender mountings. In some cases such energy-absorbing elements may be needed and/or high strength steel dolphin piles may be needed to absorb the total energy at acceptable values of contact force.

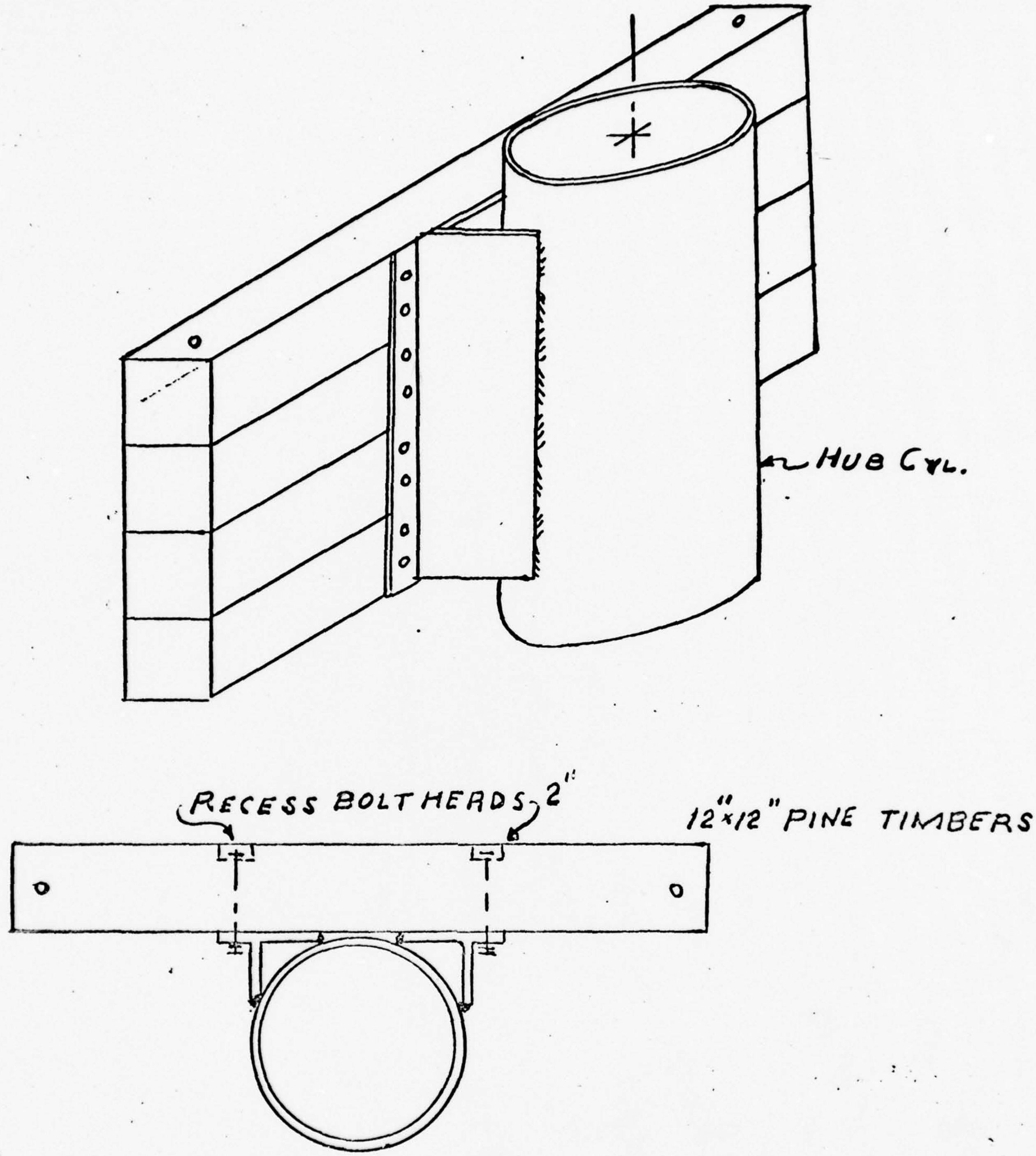
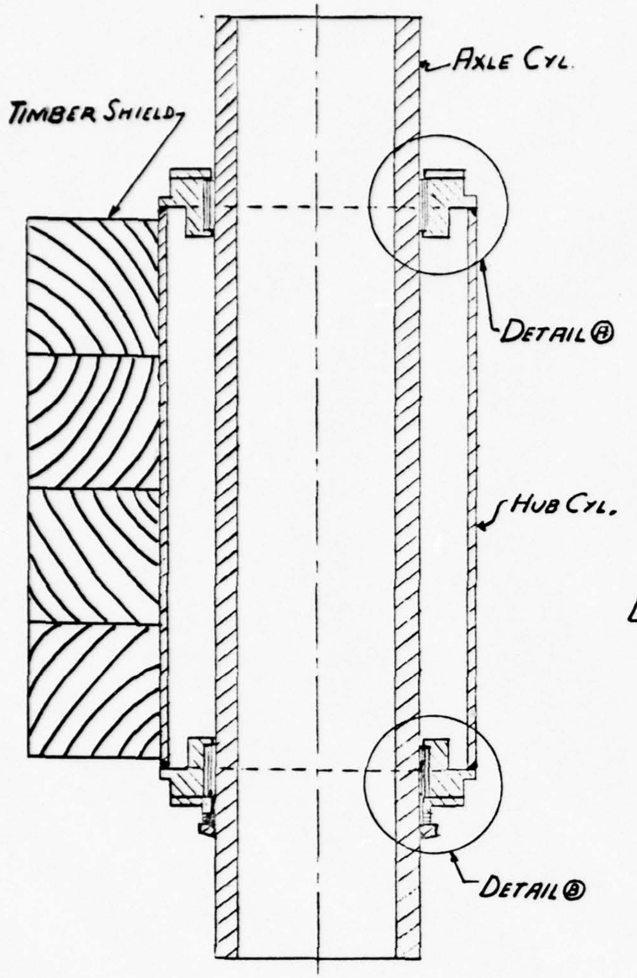
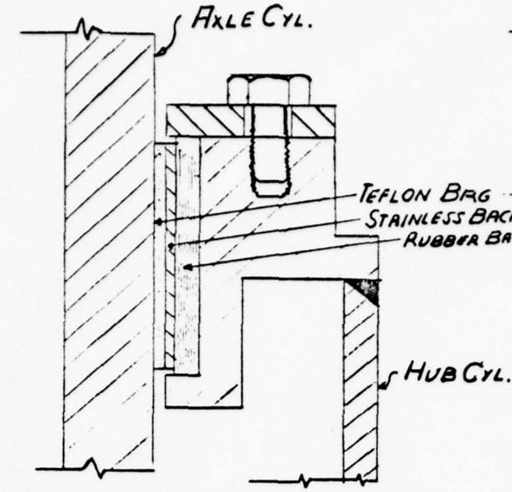


FIG 3.-SHIELD-TO-HUB-CYL. SUB-ASSEMBLY

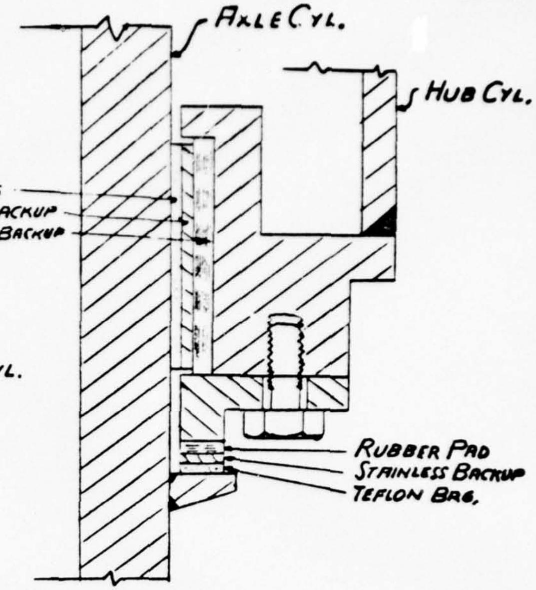
FIG. 3



Vertical Section Thru Assembly Axis
1" = 10"



Detail Ⓐ – Upper Radial Bearing
1" = 2 1/2"

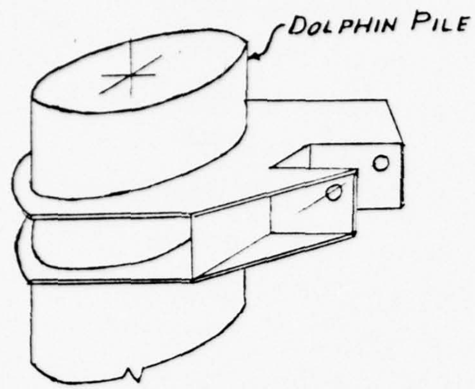


Detail Ⓑ – Lower Radial Bearing and Thrust Bearing
1" = 2 1/2"

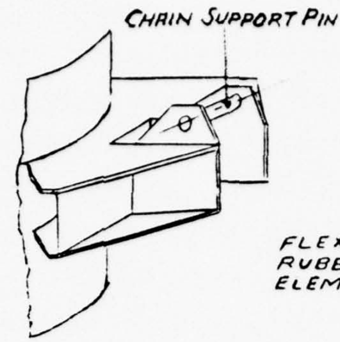
Notes

1. Same brg. details for tire-stack fenders
2. I.D. of Teflon brg rings .125" to .250" greater than axle cyl. O.D.
3. Teflon brg. rings to be sized for 3000 psi at max fender force

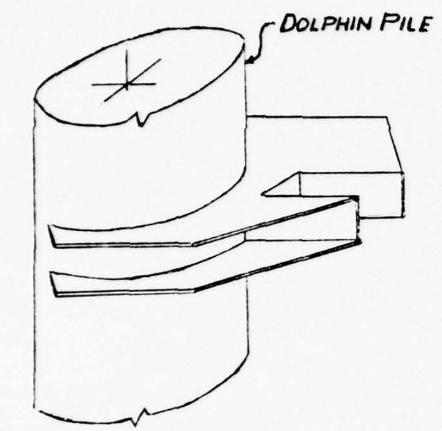
Hansen, Holley and Biggs		
Sca. 12-8-76	Approved by	Drawn by / Revised
Fig. 4-Hub-Cyl.-to-Axle Bearing Details		
		Drawing number: Fig. 4



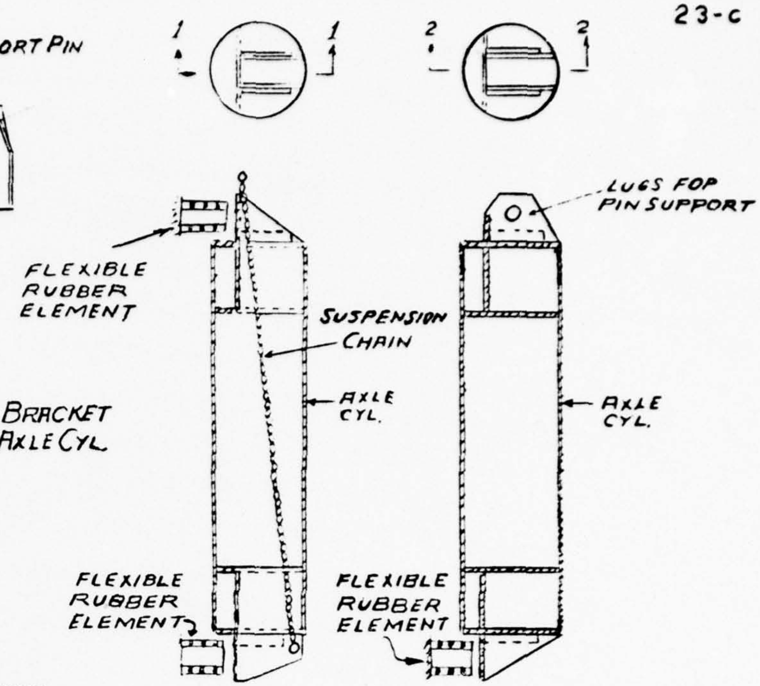
UPPER BRACKET FOR PINNED AXLE END



MODIFICATION OF UPPER BRACKET FOR CHAIN SUSPENSION OF AXLE CYL.



LOWER BRACKET



SECT. 1-1 CHAIN-SUSPENDED

SECT 2-2 PIN-SUPPORTED

NOTES

1. SHIELD FENDERS REQUIRE LOWER FLEXIBLE ELEMENTS TO ACCOMODATE PILE-TOP SLOPE; SOME TIRE-STACK FENDERS MAY BE SUFFICIENTLY FLEXIBLE TO ACCOMODATE SLOPE WITH BOTH ENDS OF AXLE PINNED.
2. CHAIN-SUSPENDED AXLE PERMITS (ENERGY-ABSORBING) FLEXIBLE ELEMENTS AT UPPER AND LOWER ENDS, WHEN REQD.
3. ENERGY-ABSORBING ELEMENTS SHOULD BE OF THE "RUBBER" "BUCKLING" TYPE, SIZED TO AVOID EXCESSIVE ECCENTRICITY OF SHIELD FORCE.

HANSEN HOLLEY AND BIGGS		
SCALE	APPROVED BY	DRAWN BY
DATE 12-8-76		REVISED
FIG 5-AXLE-CYL-TO-BRACKET DETAILS		
		DRAWING NUMBER FIG. 5

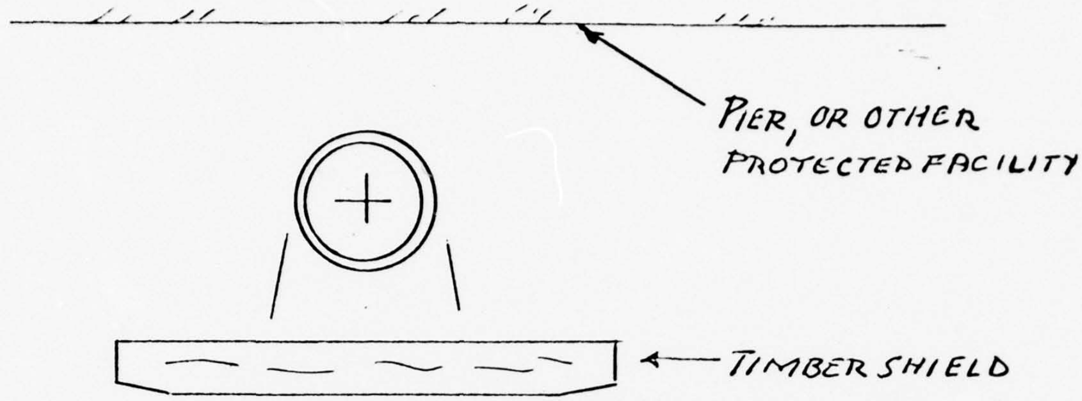
Where horizontal angular articulation is required (virtually all cases), the timber shield is mounted on a hub cylinder. This, in turn, transmits shield forces, through Teflon bushings, to an axle cylinder. The Teflon bushings can be loose fit, and they should be relatively durable in the corrosive environment. It should be emphasized that there is no functional requirement for precision in these buildings. Neither is there any need to achieve an unusually low friction coefficient. The need is only to achieve a bearing which will not have its life shortened (by binding or by destruction of its components) as a consequence of corrosion. It is believed that Teflon bushings will accomplish these purposes, will not require lubricants, and will maintain friction coefficients sufficiently low to preclude excessive vessel-to-shield edge pressures.

Where the fender shield must be permitted to move vertically to accommodate tidal changes, the axle cylinder is made enough longer than the hub cylinder to permit such movement. In all cases a Teflon thrust bearing ring transmits the vertical weight of the shield and hub cylinder and any vessel-to-shield downward vertical friction force. The Teflon bearings are backed by steel and rubber fillers, to assure satisfactory distribution of bearing pressures. Bearing components are machined to the proper dimensions and welded to the hub cylinder.

Methods of determining the slope of the dolphin axis, under loading, are presented in Reference 1. To these dolphin slope values must be added any anticipated vertical angles associated with hull form and vessel roll. Typically the total vertical angle to be accommodated will be small; i.e., less than 4 degrees. To accommodate this vertical angle, the axle cylinder transmits its principal horizontal reaction component at upper and lower ends through flexible rubber elements to the dolphin structure. The other component of horizontal reaction, at each end, is transmitted by restraining chains or, alternatively, by restraining guideways. Vertical components of dead weight, and shield friction, are transmitted from axle cylinder to dolphin structure by chains. It may be noted that the hub cylinder to axle cylinder detail is equally applicable to tire-stack fenders. See Sect. 7.

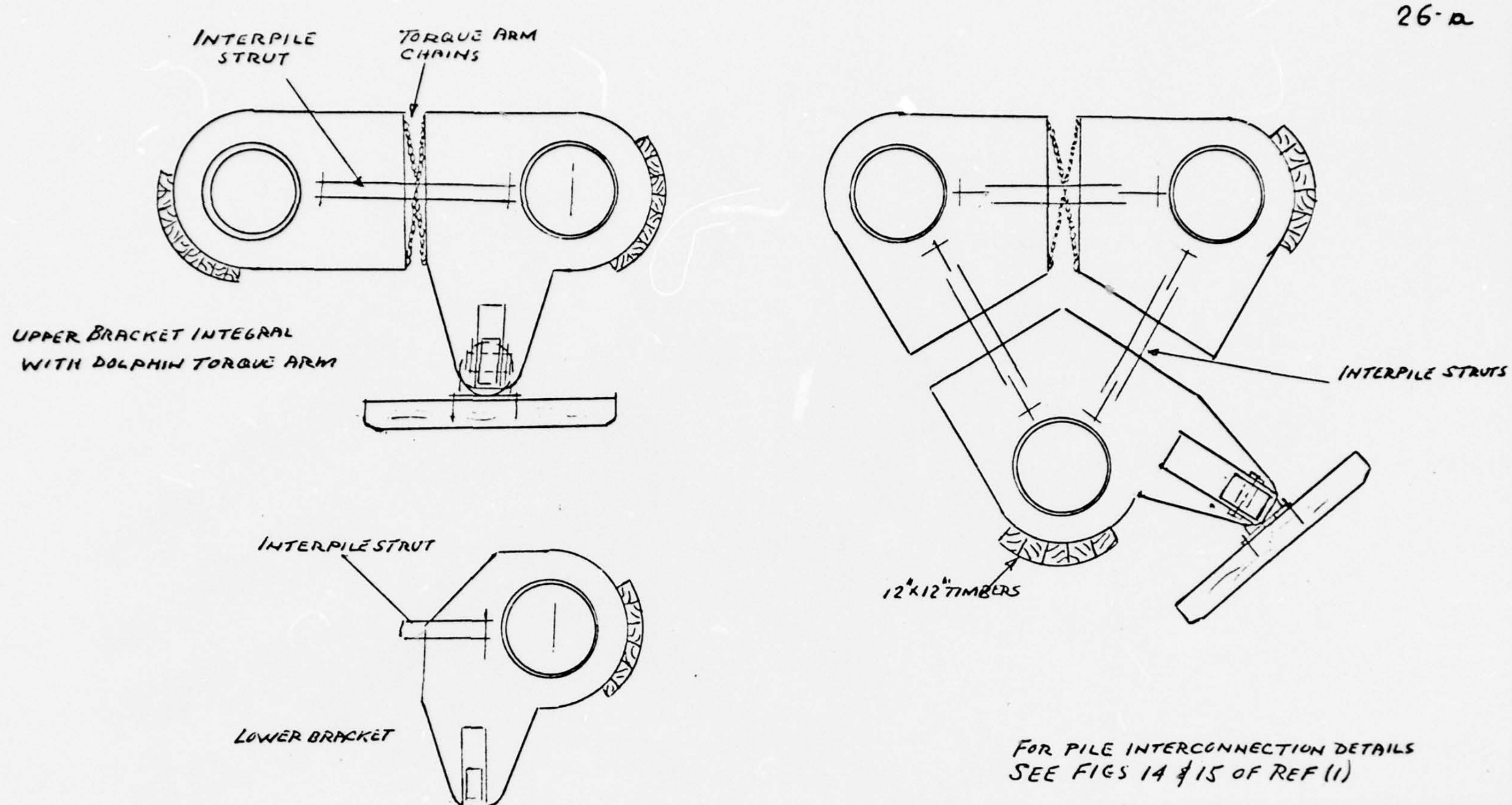
The upper point of connection of the axle cylinder to the dolphin structure is at the dolphin torque arm bracket or, in the case of a single-pile dolphin, on a special bracket arm connected to the pile. At the lower end the le cylinder is connected to a bracket extending from a pile.

Figures and 7 show typical locations of shield fenders on single-pile and multiple-pile dolphins.



1. NORMALLY THERE WILL BE ONLY ONE FENDER SHIELD PER SINGLE PILE DOLPHIN.
2. "AT REST" POSITION OF FACE OF SHIELD SHOULD BE ESSENTIALLY PARALLEL TO PIER OR OTHER FACILITY SERVED BY THE DOLPHIN
3. ROTATION OF HUB CYLINDER, ABOUT AXLE CYLINDER WILL ACCOMODATE RANGE OF APPROACH ANGLES AND HULL FORM.

FIG 6- SHIELD-FENDER LOCATIONS ON SINGLE-PILE DOLPHIN



HANSEN HOLLEY AND BIGGS

SCALE	APPROVED BY:	DRAWN BY
DATE: 12-9-76		REVISED

FIG 7.- SHIELD FENDER LOCATIONS ON MULTI-PILE DOLPHINS

DRAWING NUMBER: FIG-7

7. TIRE CASINGS FOR DOLPHIN FENDERS

7.1 Prior Applications of Tire Casings to Fenders

Used tire casings have a long history of use, as very simple fenders, or "bumpers" in the marine field. The typical applications have involved both tires draped over the contact faces of piers and tires draped over the prow and sides of small boats (particularly tug boats). In both of these common applications the tires have been relatively small in size, and the tire casings usually have been uninflated and empty. The orientation has been for loading essentially normal to the sidewalls. Because of the relatively small tire widths (i.e., face-to-face) these applications typically have involved rather limited vessel-to-pier or vessel-to-vessel relative deflections, during contact. Furthermore, when empty casings are used, only very modest resisting force can be developed before the two side walls are essentially in contact with each other. Accordingly, tires used in this way provide only limited energy-absorption. Nevertheless, their continued use, over many decades, testifies to their effectiveness in distributing contact forces over sufficient area to avoid local damage. Of much greater importance is the evidence, which this usage affords, that tire casings can withstand many cycles of severe strain, and can do so in salt water as well as in fresh water environments.

Reference (4) provides descriptions of two types of hung fenders which apply tire casings in a somewhat more sophisticated manner. In the first of these applications, ten used, empty, auto tire casings are tied together in a stack. Within the central core of the stack is placed a cylindrical canvas bag filled with "cork, old rubber pneumatic life preservers, shredded rubber tires, waste hemp, etc." Because this fender has a substantial outside diameter (approximately 30") it provides resistance over a fairly large vessel-to-pier relative displacement. Moreover, the filled core probably provides significant resisting force over a substantial portion of this displacement. Accordingly we would expect it to be fairly effective as an energy-absorbing device. However, the construction, particularly the canvas bag, appears somewhat fragile. It is not clear how successfully this

fender would resist friction force components, particularly such components acting parallel to the fender axis. It is particularly interesting to note that tires with no more than two layers of fabric worn through were considered acceptable for this fender application. Also, by implication, blowout holes were acceptable for all except the two end tires. The fender is hung, with its axis vertical.

The second type of hung fender described in Reference (4) also is comprised of a number of used, empty, auto tire casings, tied together in a cylindrical stack. However, this stack is mounted on a 16" diameter timber log turned to fit snugly inside the core space of the stack. The timber log extends out 18" beyond each end of the tire stack, and a wire rope pendant is provided near each end of the log. The fender is hung, or floated, with its axis horizontal. Because of its solid timber core this fender is less compressible than the fender having a filled canvas bag core. Nevertheless, it is reported to have excellent cushioning qualities and to be inexpensive to construct. It is reported to be very durable. For this application, as for those previously discussed, <u>used</u> tire casings are perfectly acceptable. Tread wear is of no consequence and, in some cases, it may be advantageous. That is, a smooth casing perimeter may produce lower friction components of the contact force.

For several years the Firestone Burleigh Co. Ltd., of Brentford, England, has marketed two types of pneumatic marine fenders, incorporating tires as the principal elements. The first of these, termed a "floating penumatic fender" is comprised of two or more large-diameter tires fitted to standard rims which are mounted on a common tubular axle so that each wheel is free to rotate independently. The tires are inflated and the entire fender unit is buoyant. Thus in use the fender floats, with its axle horizontal, between vessel and pier. When squeezed by a contacting vessel, major deformation of the tire occurs both in the zone contacted by the vessel and in the zone contacting the pier face (or second vessel). Thus substantial energy absorption is effected through the displacements associated with deformations in both of these zones. The tires are large to <u>very</u> large; i.e., diameters ranging from 54" to 114" in the five standard fender sizes. Zero-

load tire pressures range from 50 psig in the 54" tires to 80 psig in the 114" tires. Minimum length is of the order of 15 feet; thus they are appropriate for use between vessels and straight, solid-faced, piers, but would require some modification for use as dolphin fenders. The tires in this application differ from those of the previously discussed applications in the following significant aspects.

a) New, large tires are used. They are of a type produced principally for earthmoving equipment, and they have treads which presumably are dictated by that usage.

b) They are inflated. This fact provides the casings with sufficient strength to resist large friction force components; i.e., components parallel to the fender axle. Furthermore, the inflated tires are sufficiently buoyant to permit the entire fender assembly to function as a floating unit.

c) Since the vessel comes in direct contact with the tires, contact pressure on the hull cannot be less than the no-load tire inflation pressure (50 - 80 psig). At large loads there is a reduction in tire volume, which must increase the pressure. We estimate that final pressures probably do not exceed no-load pressures by more than ten percent. Still it must be noted that even the initial pressures are larger than the values of fender-to-hull contact pressures which the Navy has specified to be acceptable.

The second type of pneumatic marine fender developed by Firestone Burleigh Co. uses inflated tires of the same kind, and same sizes, as are used in the above-described floating fender. One or more of these large-diameter tires, fitted to rims, are mounted on a common vertical axle. The upper and lower axle bearings are mounted in guideways which permit the assembly of axle and tires to move freely back against a pair of reaction rollers. Thus, in action each tire is loaded (by the vessel) at one zone of its perimeter, and this load is balanced (by the reaction rollers) at two other zones on its perimeter. The entire assembly is housed in a steel

casing, which is installed at a fixed elevation on the pier or other protected shore facility. The major feature of this type fender is said to be the ability of the tires to rotate, even under maximum load. However, it can hardly be said that this is "free" rotation. Forces to be transmitted to the support structure include horizontal and vertical friction components which can be of the order of 50 - 60 percent of the principal component (i.e., normal to the face of the berth). The tire-related features are essentially the same for this fender as for the previously-described floating fender. These features include the use of new, large tires produced primarily for earthmoving vehicles, and the use of inflation pressures sufficient to strengthen the tire casing against force components parallel to the supporting axle. Again the tire-to-vessel contact pressures exceed the limiting values prescribed by the Navy for the present study.

Very innovative applications of tire casings to marine fenders have been achieved by the Toronto Harbour Commission. In all cases these applications utilize large-diameter (i.e., 6' ±), used tire casings, which are not inflated but are stuffed with shredded rubber. Three of these applications which have been in service for several years are described below. Drawings of these devices provided through the courtesy of Mr. Jack H. Jones, Chief Engineer of the Toronto Harbour Commission, are not included herein, but are submitted separately for study.

a) Draped, Single Casings

This is simply the common practice of draping used casings against the face of a solid pier. The innovation is in the use of large-diameter casings stuffed with shredded rubber. These simple fenders are very extensively used along the waterfront facilities managed by the Toronto Harbour Commission. They are reported to provide very satisfactory, essentially maintenance-free, service.

b) Pier Corner Fender

This application is superficially similar to the previously described Firestone Burleigh fender, but of much simpler construction.

Two or more large-diameter, used, tire casings (stuffed with shredded rubber) are mounted on a common cylindrical steel hub. The hub (28" O.D. steel pipe) is mounted on, and free to rotate about, a vertical cylindrical steel axle. The axle (24" O.D. steel pipe) is supported at each end by a used tire casing stuffed with shredded rubber. Each of these bearing tires fits snugly on the axle, and the tire perimeters bear against curved seats in the structure. Chains linking the upper end of the axle to the structure, and a key at the lower end, constrain the bearing tires to stay within their supporting curved seats.

It is of interest to compare this device with the similar, more refined, Firestone Burleigh fender. Because the Toronto casings are not infated, they probably exert smaller hull contact pressures, particularly at partial load. At maximum loading, however, the Toronto fender may exert higher hull contact pressures than does the Firestone Burleigh fender. We would expect the Toronto fender to develop smaller friction force components. The horizontal friction force component should be smaller because the perimeters of the main tires contact only the vessel; thus friction forces resisting rotation occur not at the large radius of the tire perimeter, but at the smaller radius of the supporting axle. The vertical friction force component should be smaller because the stuffed tires provide less resistance to vertical deflection of the vessel-to-fender contact zone than do inflated tires. The Firestone Burleigh fender is available with substantially larger tires and, correspondingly, greater capacities than the Toronto fender. Unfortunately it is difficult to estimate the capacity of the latter. Finally, it must be noted that the Firestone fender mechanism has been engineered to function in a salt water or fresh water environment. In contrast, the Toronto fender functions in a fresh water environment. The simple (loose-fit, steel-to-steel) bearings, which are reported to require very little maintenance, might be very short lived at a salt water site.

c) Dolphin Fenders

Entrances to ferry docks are provided with systems of single-steel-pile dolphins. These dolphin piles typically are 30" diameter or less.

Several large-diameter, used tire casings (stuffed with shredded rubber) are fitted over the upper ends of these piles to serve as fenders. In some applications the tire casings are fitted to a separate cylindrical "hub" which is of larger diameter than (and fits over) the dolphin pile. The hub is provided with simple bearings which permit it to rotate freely about the pile axis, thus minimizing friction components of the vessel-to-fender contact force. The experience with these tire casing fenders is reported to be very satisfactory.

The Toronto Harbour Commission has plans for additional innovative applications of large-diameter, used, tire casings. One particularly interesting design is for a single-pile dolphin supported in a steel caisson sleeve. Transfer of forces from pile to sleeve will be through "bearings," each comprised of three or four tire casings. The flexibility of these bearings will provide much larger displacements of the dolphin loading point than could be achieved by pile flexure and soil strains only.

Mr. Jones, Chief Engineer of the Harbour Commission, advised us that they have been able to construct the described examples of fendering at far less cost than would have been associated with alternative designs that did not exploit large, used tire casings. To date the cost of the used casings themselves has been as little as $1.50 per casing. [Moreover, in many sections of the United States, owners of used tires have had to pay for their disposal; that is, they are available essentially for the cost of transportation.] In Toronto the casing stuffing operation, and much of the fender construction and installation, is done by the work forces of the Harbour Commission.

From the above-described prior experience with tire casings as fender elements, the following points appear to us to be most significant.

a) There is abundant evidence that tire casings are tough elements which can endure repeated severe strains without loss of function, even in the harsh salt water environment.

b) Inflated tires are an attractive means of achieving floating fenders, and the Firestone Burleigh experience with floating pneumatic fenders indicates that it is feasible to maintain the no-load inflation pressure without frequent re-inflation.

c) Either inflated tires or stuffed tire casings can serve as effective "journal bearings" to accommodate small angular displacements. This is particularly important because most conventional bearings are difficult to maintain in the marine environment.

d) Vessel-to-tire friction coefficients can be very large, particularly if the contact zone is dry; submerged contact may develop a much lower friction coefficient. If the tire does not rotate about a vertical axis, to accommodate the horizontal tangential component of hull motion, the horizontal friction component of contact force may be unacceptably large. If tire rotation to relieve this force component cannot be provided, it may be necessary to interpose a shield between vessel and tire. Vessel-to-shield friction coefficients can be substantially less than vessel-to-tire friction coefficients.

e) Air-inflated tires probably are stiffer than stuffed tire casings with respect to a friction force component parallel to the tire axis. Thus hull-to-tire relative vertical displacement may produce a larger vertical friction force component on an air-inflated tire with its axis vertical than the corresponding force component on a similarly-oriented stuffed tire casing. The relative vertical displacement (due to vessel heave or roll and to dolphin axis inclination associated with pile bending) typically would be small; however, it should be considered if air-inflated tires are used.

f) Tires loaded on their side walls (normal thereto) present a larger contact area than tires loaded on their perimeters. However, for maximum compression of a side loaded tire casing it may be necessary to omit any metal rim. If the casing is air-inflated, it is essential to use a tube to preclude air leakage. If loading is normal to the side

walls, and the metal rim is omitted, a flexible throat closure of some kind will be needed to contain the tube (or shredded rubber stuffing).

g) Firestone Burleigh pneumatic fenders use tires at no-load inflation pressures which imply hull-to-tire contact pressures much in excess of the maximum acceptable hull pressures specified by the Navy for purposes of the present study. If air-inflated tires are used in contact with the vessel, they must have no-load inflation values consistent with the prescribed contact pressure maxima. However, air-inflated casings not in contact with the vessel may be inflated to higher no-load pressures, i.e., up to the tire rated pressure.

7.2 Tire Casing Functions in Dolphin Fenders

As was pointed out in Section 5.0, the fender contact element and/or the connection details must accommodate displacement and rotation components of the contact zone on the vessel, relative to the dolphin structure. Because of the large distortions which can be imposed on tire casings, such casings may be used to provide much, if not all, of the required articulation.

The large strains (shape changes) which can be imposed on tire casings imply large elastic energy absorption capacity. Where large tire casings are used, they provide supplementary energy absorption sufficient to meet the most demanding conditions.

A fender configuration utilizing tire casings, designed primarily to achieve the necessary articulation, may be an effective energy-absorbing device as well. If this is the case, it may be feasible to use a lighter dolphin structure than otherwise would be required.

7.3 Tire Fender Sub-Assembly

Steel dolphins may be of either single-pile or multiple-pile construction. Multiple-pile dolphins may be any of a variety of different configura-

tions (e.g., as presented in Reference (1)). Depending upon the site and upon the dolphin purpose, loading may be imposed only at a single location on the dolphin perimeter, or at several locations. It appears desirable to design a tire fender sub-assembly which, with limited modifications, can be applied to all steel dolphin configurations.

Fig. 8 shows a tire fender sub-assembly which should be applicable to a wide range of conditions. It is, essentially, the fender assembly developed by the Toronto Harbour Commission, with minor modifications to provide for the more corrosive ocean environment. These modifications include substitution of Teflon bearing rings for metal-to-metal bearings, and the use of greater thicknesses in the steel cylinders. The main features are:

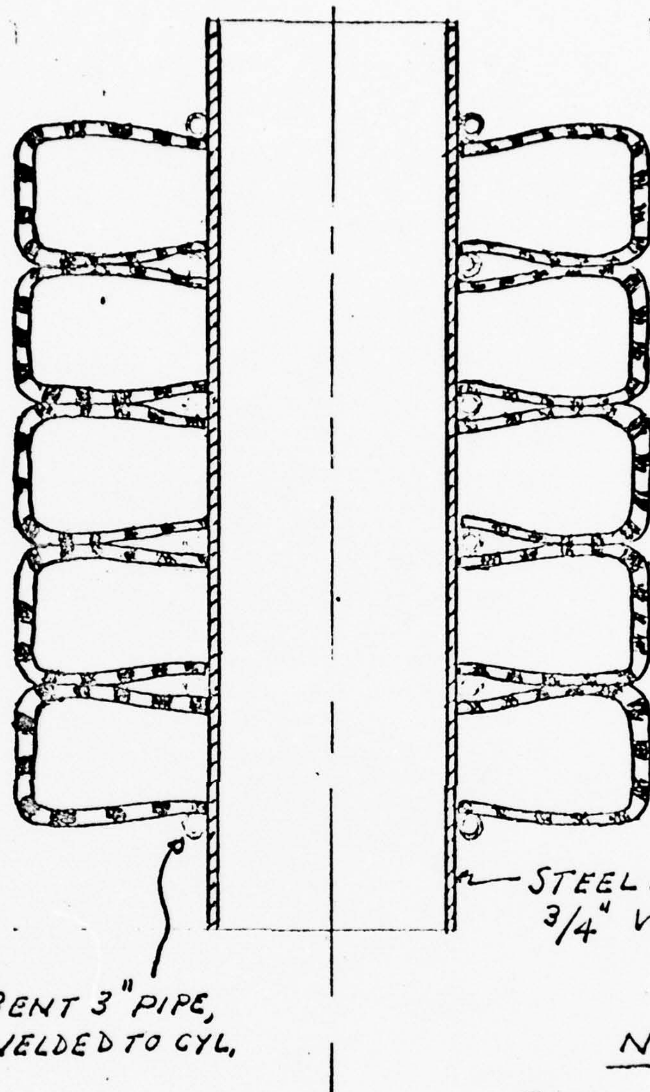
(a) A stack of (used) tire casings mounted upon a common steel hub cylinder, with the stack axis essentially vertical.

(b) Loose fitting Teflon bearing rings to permit free rotation of the hub cylinder about the supporting axle cylinder.

(c) Teflon thrust bearings to restrict axial displacement of the hub cylinder, relative to the axle cylinder, for those applications in which the tire stack and hub cylinder are not permitted to float with tidal changes.

(d) Foam flotation cylinder (or foam blocks) to support the weight of tire stack and hub cylinder in those applications in which accommodation to tidal changes is required.

It should be noted that the tire casings may be either stuffed with shredded rubber, as in the Toronto Harbour Commission installations, or may be inflated. If inflated, the casings must be provided with tubes. The advantages of stuffed casings are (i) more severely worn casings can be used, whereas inflated casings must be in sufficiently good condition to resist inflation without danger of blowout, (ii) complete freedom from concern regarding possible loss of inflation pressure, (iii) direct application of a concept which has been proven through several years of prior successful application. The advantages of inflated tire casings are (i) buoyancy, which will permit flotation of the stack without the necessity for foam flotation elements, (ii) availability of reliable data on load/deformation characteristics.



NOTE

ARRANGEMENT SHOWN, FOR TIRES STUFFED WITH SHREDDED RUBBER, HAS BEEN USED BY TORONTO HARBOR COMMISSION

FIG 8 - TIRE STACK-TO-HUB-CYL ASSEMBLY

The Firestone Burleigh "floating pneumatic fender" might be modified for use as a vertically oriented, free-rotating, tire fender sub-assembly. This modification would require the use of a unit with intermediate tires of larger diameter than the two end tires. The smaller end tires would be fastened to the dolphin, to serve as flexible end bearings, similar to the detail used by the Toronto Harbour Commission. However, the present axle used in the Firestone Burleigh "floating pneumatic fender" might not be strong enough to withstand the bending moments which would be developed in the described modification. Further, the modified unit could not easily be further modified to accommodate tidal changes.

7.3.1 Provision for Tire Stack Rotations

The decision to provide for "free rotation" of the tire stack about the stack axis reflects considerable study of alternative arrangements. It was influenced by (a) the very large horizontal friction force component developed by the Firestone Burleigh pneumatic fender, (b) the successful experience reported by Toronto Harbour Commission with tire casing fenders incorporating very simple provisions for rotation, (c) the desirability of avoiding the imposition of unnecessarily large horizontal friction force components on the dolphin structure or on the vessel. The decision requires, of course, that the tires be loaded at one point on their perimeters only, that is, at the vessel-to-fender contact zone, with the equilibrating reaction provided by the steel rim cylinder. In this state the energy absorption is only 50 percent as much as if both the load and the reaction were applied on the perimeter (e.g., as in the Firestone Burleigh floating fender). The reduced resistance to stack rotation, and greatly reduced friction force, are of greater importance and, in our judgement, justifies the choice.

7.3.2 Provision for Free Axial Displacement of Tire Stack

It may be necessary or, at least, desirable to provide for free axial displacement of the tire stack and hub cylinder. The most obvious reason for providing such freedom is to permit accommodation to tidal changes. In addition, however, dolphin axis slope changes, slope changes of the (flexibly supported) axle cylinder, vessel heave, and vessel roll, all may tend to cause small axial displacements. Freedom of hub-to-axle relative displacement

will minimize axial forces associated with such effects. These forces are not beneficial, and they could shorten the service life of the assembly. Loose fitting Teflon bushings will permit free axial displacement as well as free rotation.

7.3.3 Mounting of Tire Casing Stack Sub-Assemblies on Single-Pile Dolphins

Ref. (1) provides single-pile steel dolphin designs for pile diameters of 36" and larger. For water depths and vessel classes of interest, smaller diameter piles will rarely, if ever, be economically attractive. (See, however, Section 8.3.2). It is highly desirable to use as few piles as possible to eliminate, or to minimize, the required inter-pile structural framing. By using larger pile diameters and wall thicknesses, and particularly by using higher strength steels, single-pile dolphins will be applicable to a large part of the range of required dolphin energies.

For the large pile diameters of interest, only the largest earthmoving vehicle tire casings would be large enough to fit directly on the dolphin pile. If purchased new, these large tires would be very expensive. As used tires, their large size would limit availability. Moreover, there might be no tires, new or used, large enough to fit over thelargest dolphin piles of interest.

Although the Toronto Harbour Commission has successfully constructed single-pile dolphins with the tire casings hub cylinder mounted directly on the pile, it must be noted that these are dolphin piles of relatively small diameter, in relatively shallow water depths. Accordingly, the dolphin pile deflections are small. When the tire casings compress and the dolphin deflects, the vessel does not risk contact with the lower (unprotected) part of the pile. In contrast, the water depths here of interest (40' - 70') are greater, and the deflections are substantial. For this reason it does not appear to be practical to mount the hub cylinder directly on the dolphin pile, using the latter as the axle cylinder. A further reason for not using such a detail is that it does not permit any means of articulation of the hub cylinder to accommodate to slope changes of the deflected dolphin axis.

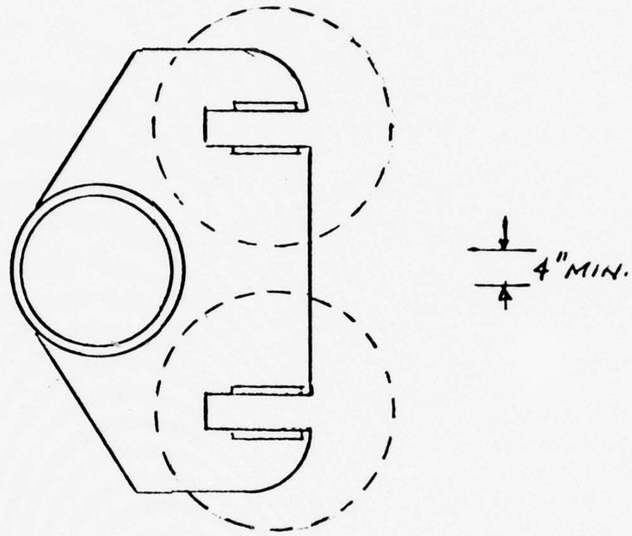
For the above reasons it is necessary to use independent axle cylinders, mounted at their upper and lower ends on horizontal bracket arms extending out from the dolphin pile. The bracket arms must be long enough to avoid

contact of the rotating tire stack with the dolphin pile. This requirement will assure that the tire stack is sufficiently outboard of the dolphin pile to preclude contact of the lower part of the vessel hull with the unprotected pile. Axle cylinder to bracket connections for tire stack fenders are the same as for shield fenders (Fig. 5). Brackets for common mounting of double tire stacks are shown in Fig. (9).

Fig. (10) illustrates single-pile dolphins with single and with multiple tire casing stack fenders. Note that multiple stacks may be used either (a) to accommodate vessels approaching at different points on the dolphin perimeter, or (b) to provide additional vessel-to-casing contact area. In the first case, shown in Fig.(10), each support bracket is fastened rigidly to the dolphin pile. In the second case, the bracket arms are mounted on a common cylinder which is free to rotate on the dolphin pile to accommodate variations in vessel angle of approach and variations in hull form.

When it is necessary to reduce the outboard projection of the vessel-to-casing contact point, the diameter of the upper portion of the dolphin pile may be reduced, as shown in Fig. (10). If vessels approach on one side of the dolphin only, the outboard projection of the contact point may be be further reduced by offsetting the axis of the small diameter portion of the dolphin from the axis of the large diameter portion, as shown in Fig. (10).

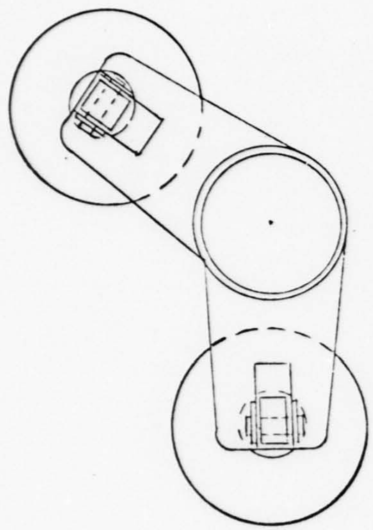
If the length of the tire casing stack is short, say 10' or less, changes in slope of the dolphin axis, relative to the vessel hull, may be accommodated by the flexibility of the casings. In such cases the connections of axle cylinders to bracket arms need not provide articulation. If articulation is required, the axle must be permitted to move radially with respect to the bracket arm. Flexible rubber elements transmit the radial component of force. When additional energy absorption is required, these rubber elements, chosen from among commercially available rubber fender elements, will provide such additional energy.



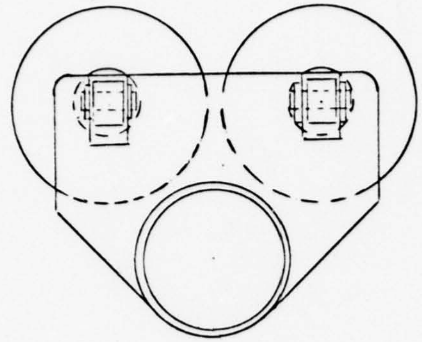
1. BRACKETS MAY BE WELDED TO THE PILE. ALTERNATIVELY, BRACKETS MAY BE WELDED TO A COMMON CYLINDER WHICH ROTATES ABOUT PILE (ACTING AS AN AXLE CYL.)

2. RECTANGULAR SLOTS RECEIVE ENDS OF TIRE-STACK AXLE CYLINDERS, AND ENERGY-ABSORBING UNITS IF REQD. (SEE FIG 5.)

FIG 9 - BRACKET FOR COMMON MOUNTING OF DOUBLE TIRE-STACK FENDERS



INDEPENDENT TIRE-STACK FENDERS
ON COMMON DOLPHIN PILE



DUAL TIRE-STACK FENDER ON
COMMON BRACKETS TO DOLPHIN PILE

NOTES

1. WHEN DUAL TIRE-STACK FENDERS ARE MOUNTED ON COMMON BRACKETS THESE BRACKETS CAN BE FASTENED TO A COMMON CYLINDER WHICH IS FREE TO ROTATE ABOUT THE DOLPHIN PILE. TO FACILITATE SUCH AN ASSEMBLY IT MAY BE NECESSARY TO REDUCE THE DIAMETER OF THE PILE IN THE REGION OF THE BRACKETS.
2. IF TIRE STACKS ARE SHORT, FLEXIBLE ELEMENTS IN THE AXLE CYLINDER SUPPORTS (TO ACCOMODATE PILE SLOPE) MAY NOT BE NEEDED. TO ACCOMODATE PILE SLOPE AN ELEMENT IS USED ONLY IN THE UPPER END AXLE CYLINDER SUPPORT. IF ADDED ENERGY ABSORPTION IS REQUIRED, FLEXIBLE ELEMENTS MAY BE USED AT UPPER AND LOWER SUPPORTS
3. SEE FIGS 12 & 13 FOR BRACKETLESS MOUNTINGS.

HANSEN, HOLLEY AND BIGGS

SCALE:	APPROVED BY:	DRAWN BY
DATE: 12-8-76		REVISED

FIG 10 - TIRE-STACK FENDER LOCATIONS ON SINGLE-PILE DOLPHINS.

DRAWING NUMBER: FIG 10

7.3.4 Mounting of Tire Casing Stack Sub-Assemblies on Multi-Pile Dolphins

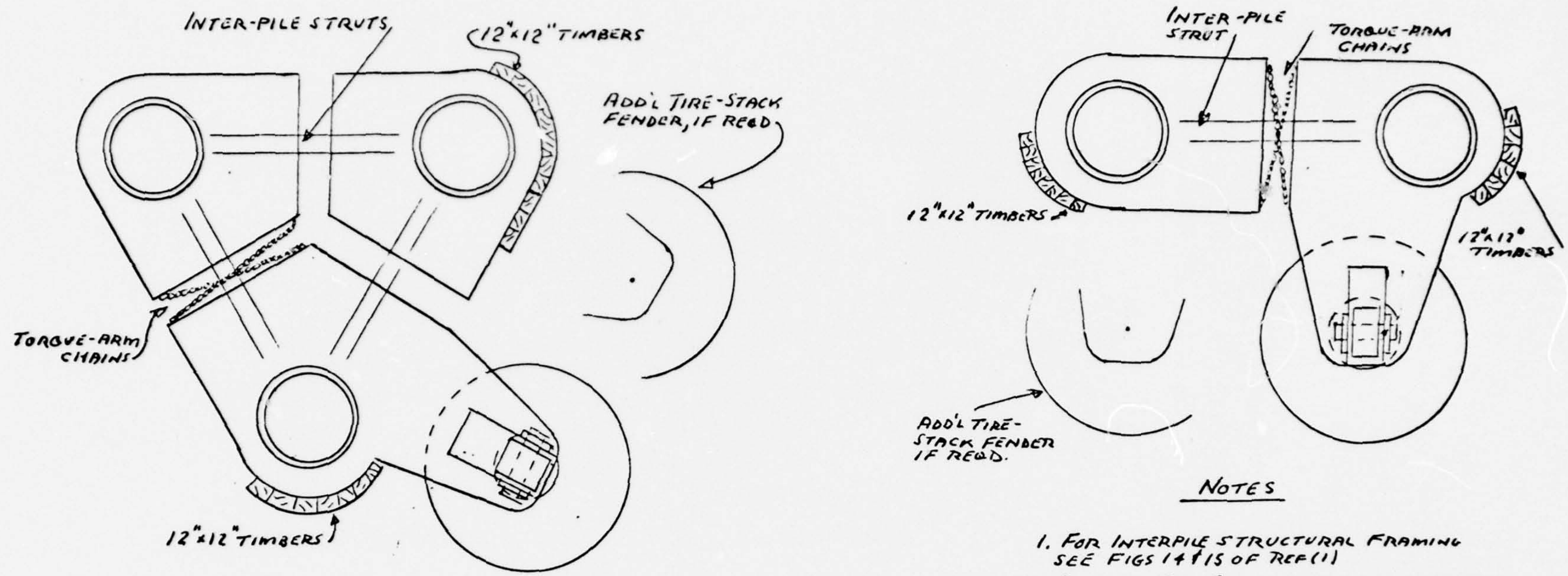
Multi-pile dolphins, presented in Ref. (1), are provided with torque arm brackets at the top. The upper fender brackets can be connected directly to these torque arm brackets. The lower fender brackets can be cantilevered directly from the dolphin piles. Tire casing stacks can be mounted at any point(s) on the dolphin perimeter, limited only by two considerations. First, excessive eccentricities of dolphin loadings. This is not a serious constraint because of the negligible friction component of vessel-to-casing force. Second, the tire stack(s) must be so located as to preclude vessel contact with the torque arm brackets, over the range of possible approach angles. Suitable arrangements are shown in Fig. (11).

7.4 Force and Energy Capacities of Individual Tire Casings

7.4.1 Inflated Tires

The maximum force which can be imposed on the tire perimeter is approximately equal to area of the loaded zone "footprint" multiplied by the internal air pressure, at maximum displacement. The internal air pressure will not exceed the no-load inflation pressure by more than perhaps five percent; conservatively, we may take the internal pressure as equal to the inflation pressure. Tires come in many shapes, sizes, and wall constructions; the area of the loaded zone (a function of tire shape) will vary from one tire construction to another.

Data on the dimensions and capacities of the tires used on the Firestone Burleigh Floating Wheel Fenders (Reference 7) may be used as a basis for estimating capacities of tires in the same general size range, but at lower inflation pressures. For this purpose one may examine the characteristics of the three smallest fender wheel tires, as presented in Reference (7); i.e., tires of 78", 69", and 54" outside diameters. Table 8 lists, for each of these tires, diameter, no-load casing width, maximum deflection, maximum load, maximum energy, and inflation pressure. For purposes of the present estimates it can be assumed that neither the maximum acceptable tire deflection nor the area of the loaded zone is a sensitive function of inflation pressure. Thus maximum load and maximum energy each is assumed proportionate to inflation pressure.



NOTES

1. FOR INTERPILE STRUCTURAL FRAMING SEE FIGS 14 & 15 OF REF (1)
2. ROTATING TIRE-STACKS DEVELOP ONLY SMALL VESSEL-TO-FENDER FRICTION FORCE; THUS SMALL DOLPHIN TORQUE. THEREFORE A WIDE CHOICE OF FENDER LOCATIONS ON PERIMETER

HANSEN, HOLLEY AND BIGGS		
SCALE:	APPROVED BY:	DRAWN BY
DATE: 12-9-76		REVISED
FIG 11 - TIRE-STACK FENDER LOCATIONS ON MULTI-PILE DOLPHINS		
		DRAWING NUMBER FIG 11

Table 9 gives the areas of the loaded zones, at maximum deflection, for each of the three tires. These areas are simply maximum loads divided by inflation pressures. Table 9 also gives the square feet of contact area per foot of tire stack length, based on an assumed center-to-center tire spacing 1.5 times the no-load tire width. The contact areas per foot of tire stack length may be used to determine the required total length of stack (which can be provided in one or more parallel rows), corresponding to a maximum applied dolphin load.

Table 8 - Characteristics of Tires Used on Firestone Burleigh Floating Wheel Fenders

Tire Dia.	No-Load Width	Max. Defl.(1)	Max. Load	Max. Energy(1)	Infla. Press.	F/W
(ins)	(ins)	(ins)	(kips)	(kip-ft)	(psi)	k/k-ft)
54	24.0	8.0	46.6	13.7	50.	3.4
69	27.1	7.8	70.4	19.5	60	3.6
78	30.1	10.6	171.6	72.2	80	2.4

(1) Maximum deflection and maximum energy each taken as 50% of values tabulated in Reference (7), because the latter are for equal forces on opposite sides of the tire perimeter.

Table 9 - Effective Contact Areas of Tires Listed in Table 8, at Max. Load

Tire Dia.	Contact Area	Assumed Spacing	Contact Area Per Ft. Stack
(ins)	(ins)	(ins)	(ft^2/ft)
54	932	36	2.2
69	1173	40.6	2.4
78	2140	45.2	4.0

7.4.2 Stuffed Tire Casings

It is reasonable to assume that stuffed casings can accept the same maximum deflections as inflated tires of the same size and shape. Thus it also can be assumed that contact areas per foot of stack length will be essentially the same for stuffed casings as for air-inflated tires. As the air-inflated tire goes from no-load to maximum load (i.e., zero contact area to maximum contact area) the internal pressure remains essentially constant. In contrast, the pressure in a stuffed casing increases from zero to maximum with increasing load, deflection, and contact area. Tests will be required to indicate the stuffing pressures, as a function of tire geometry and deflection. There is little doubt that stuffing pressures at least as high as the allowable contact zone pressures can be developed by stuffed casings. It may be that these pressures are developed at deflections and contact areas somewhat larger than for inflated tires of equal size. If so, this would be advantageous; however, only tests will resolve this uncertainty.

7.4.3 Force/Energy Ratios for Tire-Fendered Dolphins

The tire casings, energy-absorbing rubber elements in the axle-cylinder-to-bracket-arm connections, and the dolphin pile structure each transmit the same ship-to-fender force, F. The force/energy ratios for the dolphin structure are given in Tables 3 and 4. The force/energy ratios for (inflated) tires are given in Table 8. If energy-absorbing rubber elements are incorporated in the axle-cylinder-to-bracket-arm connections, their force/energy ratios are available from manufacturers' catalog data. These ratios can be used to assist in the selection of suitable elements to provide the added energy absorption.

W = required total energy, kip-ft.

$W_{struct.}$ = energy absorbed by the steel dolphin structure, kip-ft.

W_{tires} = energy absorbed by the tire stack, kip-ft.

$W_{mtg.}$ = energy absorbed by the rubber elements in the mountings, kip-ft.

F = the dolphin force at rated energy, kips.

$(\frac{F}{W})$ = force/energy ratio for the system, kip/kip-ft.

$(\frac{F}{W})_{struct.}$ = force/energy ratio for the structure, kip/kip-ft.

$(\frac{F}{W})_{tires}$ = force/energy ratio for the tires, kip/kip-ft.

$(\frac{F}{W})_{mtg.}$ = force/energy ratio for the mounting, kip/kip-ft.

The above ratios are useful in determining the additional energy capacity provided by tire fenders. They are also useful in the selection of energy-absorbing elements, if needed, in the mountings.

Example

A proposed dolphin structure will be comprised of three 48" piles with $t = 0.75$" and $f_y = 60$ ksi. Water depth is 60 ft. The rated energy for this pile grouping is 155 kip-ft. An array of tire fender stacks with $(F/W)_{tires} = 3.4$ is proposed. Determine the combined energy capacity of structure and fender tires. Determine, also, the required energy $W_{mtg.}$ and $(F/W)_{mtg.}$ of energy-absorbing elements in the mounting to raise the total energy capacity to 300 kip-ft.

Solution

From Table 3, $(F/W)_{struct.} = 1.22$

$$F = (F/W)_{struct.} \times W_{struct.}$$

$$= (1.22)\ 155 = 189 \text{ kips}$$

$$W_{tires} = F \div (F/W)_{tires}$$

$$= 189 \div 3.4 - 56 \text{ kip-ft.}$$

$W = 155 + 56 = \underline{211}$ kip-ft for structure and tires.

To provide a total energy capacity 300 kip-ft, the required energy in the mountings:

$$W_{mtg.} = 300 - 211 = \underline{89} \text{ kip-ft.}$$

Assume mountings at top and bottom connections of the axle cylinder share the fender force equally, and each provides 50 percent of the additional energy. Then each element must provide 45 kip-ft. of energy absorption at 189/2 = 90 kips force.

Thus select rubber elements with

$$W_{mtg} = \underline{45} \text{ kip-ft.}$$

$$(F/W)_{mtg} = 90/45 = \underline{2.0} \text{ kips/kip-ft.}$$

8. FENDERING FOR SUBMARINES

8.1 Unique Characteristics of Submarine Hulls

Fendering of dolphins to serve nuclear submarines must reflect aspects of the submarine hulls which are unique to this class of vessel. First, transverse sections of the hull are sharply curved. To illustrate, a circular hull form of 31' to 33' diameter is representative of midlength sections. Second, freeboard is approximately 3'; thus extreme outboard points on the hull are about 13' below the water surface. Third, there are stabilizing planes, and other protuberances, which must be protected.

The hulls of submarines are inherently strong. It was indicated by the Navy that fender contact pressures in excess of 24 psi are acceptable, but the actual value of allowable pressure is classified information. In view of the much higher allowable pressures given in NAVFAC DM-25, p. 25-2-38 (up to 50 psi for vessels inherently more fragile than nuclear submarines) we assume that a contact pressure of 100 psi should be tolerable.

8.2 Present Fendering Practice

Present practice in fendering for submarines includes the use of submerged (timber and steel) camels approximately 12' deep. Cylindrical rubber fender units also are used to protect submarines from damaging contact with service tenders.

The Navy provided the following additional information relevant to the problems of fendering for submarines at shore facilities.

- Most of the submarines present only their hulls, for fender contact, over most of their length, and the hull is not readily damaged by fender contact.

- The sonar bulb, on the bow end, is vulnerable. However, the beam width of this bulb is substantially smaller than the main body of the vessel, and damaging contacts with the bulk should be easily avoided.

- Docking is virtually always accomplished with tug assistance.

- The large horizontal stabilizing vane, just forward of the propeller, is vulnerable to damage.

- Shoreside personnel prefer to moor directly against pier fender piles because the camels are costly to maintain and time consuming to place. They also prefer such mooring because it permits the use of very light walkways which can be placed manually rather than by crane (as is required by the heavier walkways required for mooring against camels).

- Vessel personnel prefer to moor against camels because they consider such mooring safer, and because it facilitates underwater inspection of the hull by divers.

8.3 Recommended Dolphins and Dolphin Fendering

8.3.1 Approach Velocities and Corresponding Energies

Based upon the foregoing information, it appears unrealistic to design dolphins serving submarines for the upper-bound velocity (0.9 ft/sec) assumed in Table 2. The "average" velocity (0.6 ft/sec) should be sufficient in the design of dolphins used to assist submarines in narrow channels and basins, and this design velocity probably is very conservative for dolphins which may be used to prevent damaging contact of the vessel with shore facility structures. Accordingly the energy-absorbing demand on isolated dolphins required for guidance in channels or at approaches to docking facilities may be conservatively taken as 80 kip-ft. (See Table 2). For rows of dolphin piles which protect a shore facility, this 80 kip-ft energy value may be shared by several piles.

8.3.2 Inapplicability of Conventional Rigid Fender Shield

The timber and steel camels currently used for mooring submarines at shore facilities typically have rigid, planar, timber contact facing. We found no evidence that these facings ever have caused damage to a vessel hull, even though they result in a line loading rather than a low, distributed pressure. This is believed to reflect the fact that mooring energies are very small and/or the fact that the vessel hulls are very strong. Never-

theless we do not believe that a conventional timber fender shield is appropriate for dolphins serving submarines. It would be impracticable to provide a rigid shield surface which would conform to the vessel hull form.

8.3.3 Dolphin Piles and Fendering Parallelling a Shore Facility

We recommend parallel, single, small diameter dolphin piles for mooring purposes at piers and other shore facilities. For fendering these piles we recommend the use of free-rotating floating tire fender stacks mounted on a hub cylinder which in turn is mounted directly on the piles. See Fig. (12). The tire casings may be either stuffed with shredded rubber or air-inflated. The dolphin piles may be either cylindrical, of 24" to 26" diameter, or may be rolled steel WF sections. The latter sections are suitable because the free-rotating tire stacks will reduce torsion effects and load components parallel to the shore to negligible values. However, if WF section piles are used they must be fitted with cylindrical bearing rings. Forces will be transmitted from the hub cylinder, through Teflon bushings, to these bearing rings.

The submarine hull form will permit the use of flotation cells near the water surface, from which the tire stacks can be suspended to float submerged at the depth corresponding to the extreme beam width of the hull.

In shallow water depths, particularly in water depths less than the minimum here considered (40'), the use of high-strength steel piles may be considered. However, it is not anticipated that this will be necessary, and 60 ksi yield steel should be satisfactory.

8.3.4 Isolated Dolphins and Fendering to Assist in Maneuvering in Narrow Channels and Entrances

For this application single-pile dolphins again are sufficient. Free-rotating tire stack fenders, floating, if necessary, to accommodate tidal changes, are again recommended. However, as shown on Fig. (13), these piles must be of somewhat larger diameter than those described in the preceding section. For this reason the pile itself cannot be used as the axle cylinder for the tire stack. Rather, a separate axle cylinder is supported on the dolphin pile.

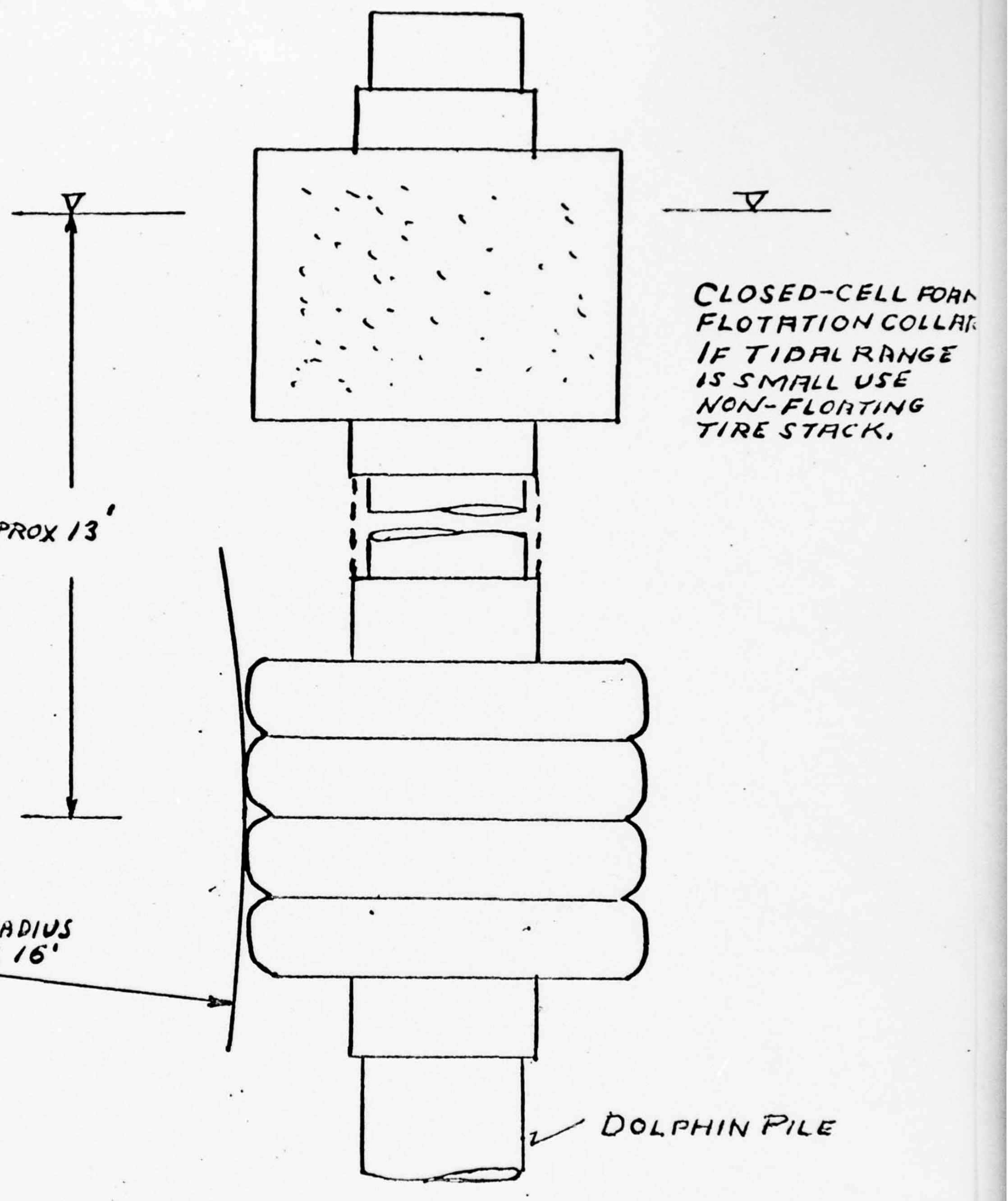
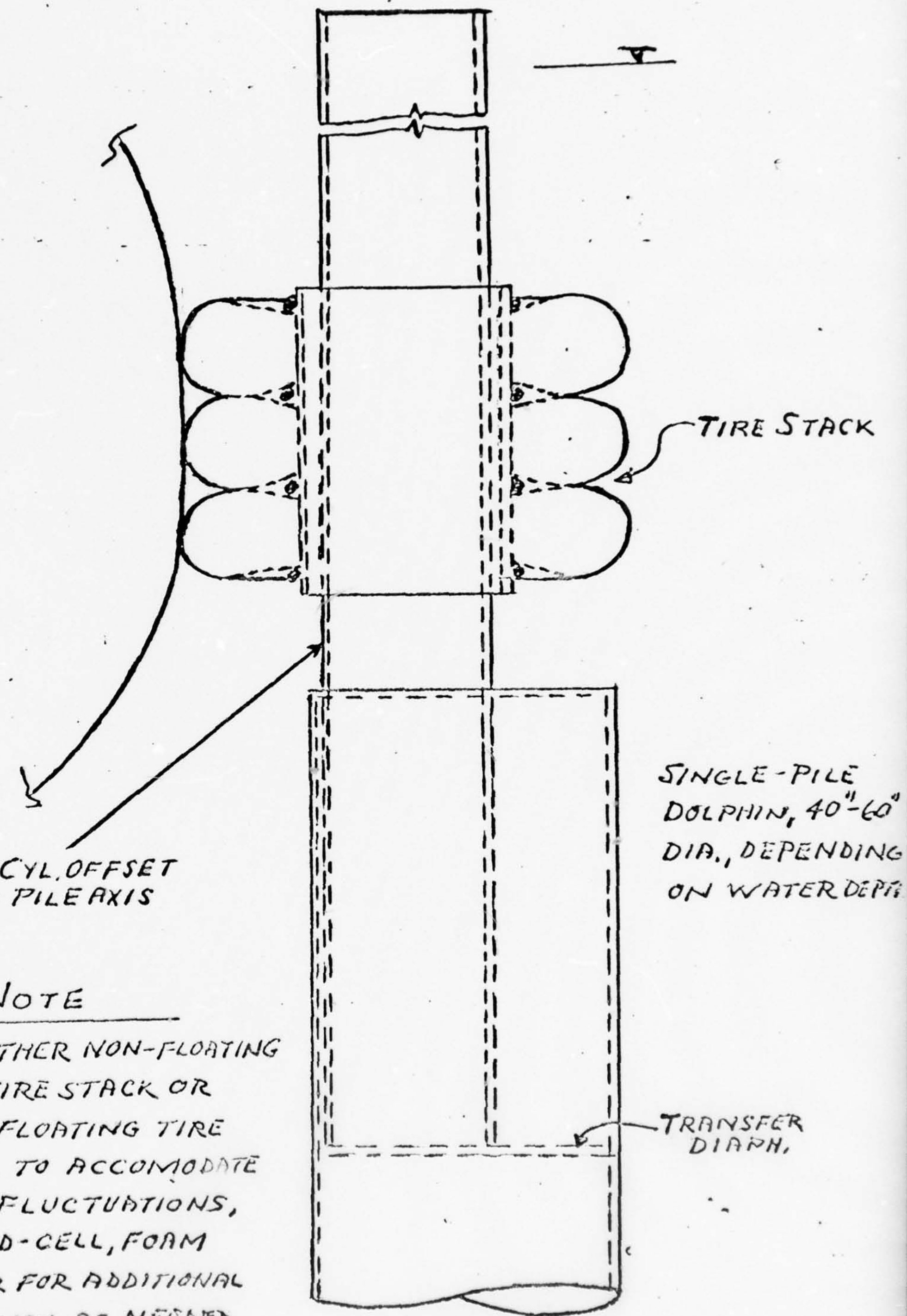


FIG. 12 - DOLPHIN WITH TIRE-STACK FENDER FOR MOORING SUBMARINES



NOTE

1. USE EITHER NON-FLOATING LONG TIRE STACK OR SHORT FLOATING TIRE STACK TO ACCOMODATE TIDAL FLUCTUATIONS,
2. CLOSED-CELL, FOAM COLLAR FOR ADDITIONAL BUOYANCY, AS NEEDED.

FIG. 13 - DOLPHIN WITH TIRE STACK FENDER TO ASSIST SUBMARINE MANEUVERING IN NARROW CHANNELS.

FIG 13

9. DOLPHIN FENDERING WITHOUT PROVISION FOR ARTICULATION

9.1 Applicability

Throughout this study we have sought to develop fender concepts which would preclude undesirable pressure concentrations on the hulls of vessels. Where conventional timber fender shields are used, this has required articulated mountings which accommodate variations in hull form, variations in angle of vessel approach, slope changes in loaded dolphins, vessel heave and roll, and tidal variations. Alternatively we have recommended tire stack fenders which, with articulated mountings, also accommodate to all of the foregoing linear and angular vessel-to-dolphin relative motions.

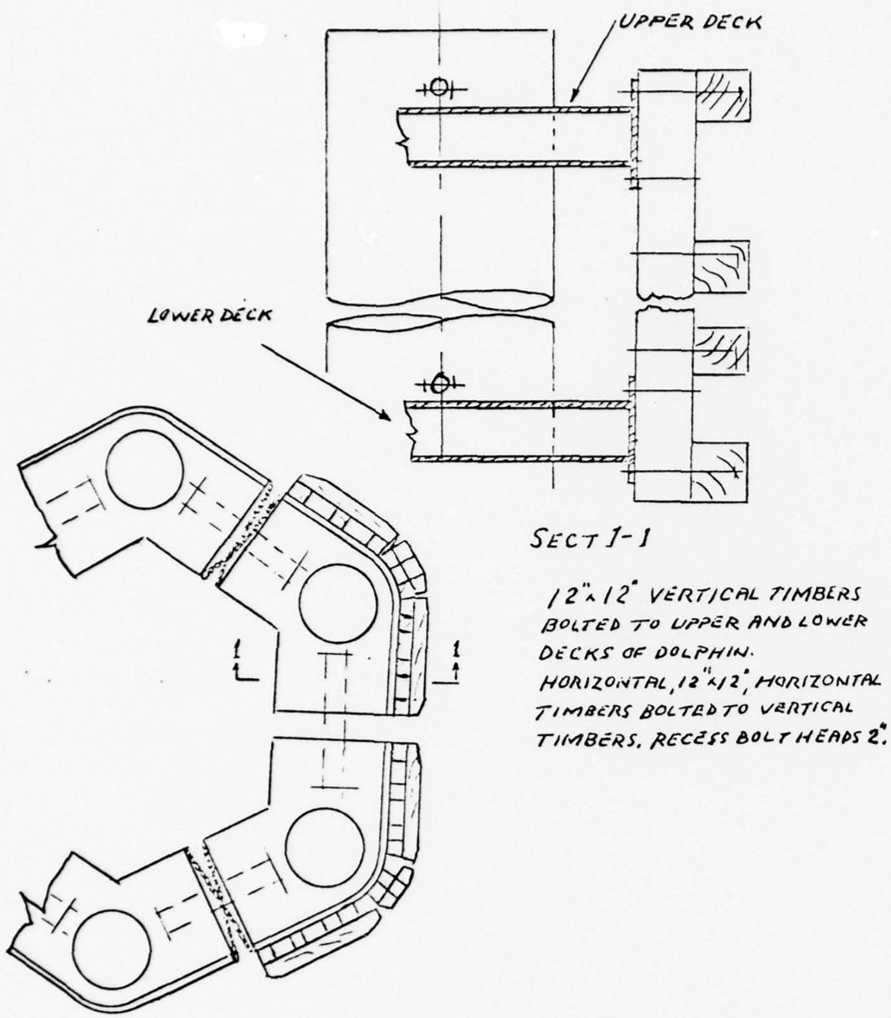
In general we have found that many of the commercially available fender elements cannot, in themselves, provide all the degrees of articulation that appear to be necessary for fendering flexible steel dolphins. Nevertheless, these commercial devices have a successful history of application, particularly on piers and on dolphins which are less flexible than are here considered. Moreover, there are many sites at which vessels are routinely docked against piers provided only with conventional fender piles, sometimes backed up (but more often not) by flexible rubber fender elements. Such arrangements reflect no attention to the vessel-to-pier load concentrations which can result if there is a significant approach angle, and they are successful, presumably, only because the approach angles are negligibly small. It must be assumed that dolphins sometimes will be employed under similar conditions.

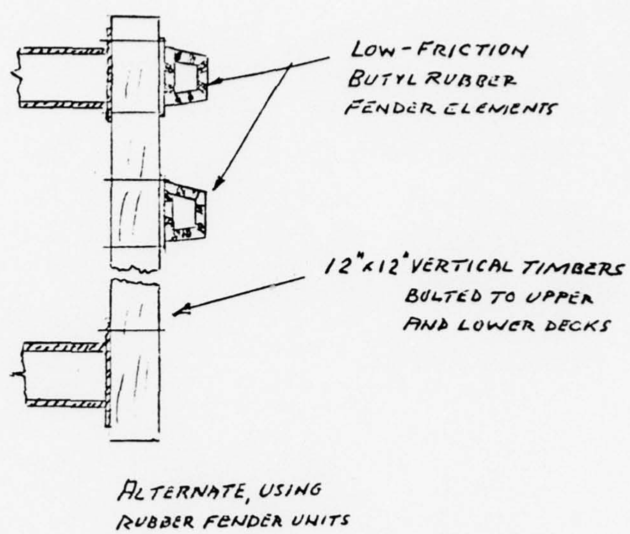
It must also be noted that there may be occasions when dolphins will be used to serve smaller vessels than those here considered (e.g., tugs, barges) which have smaller energies to be absorbed and/or can better tolerate more concentrated contact forces.

Under any of the above circumstances it may be satisfactory to use dolphin fendering which is similar to any of the fendering provisions frequently applied to piers. These include simple timber fenders, timber members backed up by flexible elements, flexible low-friction* rubber elements

* See Ref. (12) for description of low-friction Butyl rubber-fender elements and example application.

contacted directly by the hull, etc.; i.e., fender schemes which have proven to be satisfactory under conditions involving piers rather than flexible dolphins, but operationally similar with regard to approach angles, severity of exposure, vessel speed, etc. Fig. (14) shows a few fender arrangements which may be appropriate under these restricted conditions.





NOTES

(1) SEE FIGS 14 & 15 OF REF(1) FOR INTERPILE STRUCTURAL FRAMING
(2) SUGGESTED FENDERING APPLICABLE, INDEPENDENT OF NUMBER OF DOLPHIN PILES

HANSEN, HOLLEY AND BIGGS

SCALE	APPROVED BY:	DRAWN BY
DATE: 12-9-76		REVISED
FIG 14 DOLPHIN FENDERING WITHOUT PROVISION FOR ARTICULATION		
		DRAWING NUMBER FIG 14

10. INTERRELATED STEPS IN THE DESIGN OF DOLPHIN AND FENDERING

Reference (1) presents steps in the selection of numbers and sizes of dolphin piles to satisfy specified energy absorption requirements. However, the process therein outlined did not account for the energy absorption capacity of the fendering. The following modification in the selection process is required.

(a) Either assume the division of total required energy between the dolphin structure and the fendering (fenders and mountings) directly, or assume force/energy ratios, F/W, for the dolphin and the fendering. Based upon these ratios the energy to be absorbed by the dolphin structure is the required total energy multiplied by the following ratio:

$$\left| \frac{\dfrac{1}{(F/W)_{struct.}}}{\dfrac{1}{(F/W)_{struct.}} + \dfrac{1}{(F/W)_{fendering}}} \right|$$

Note that the force/energy ratio for the dolphin piles is a function of pile diameter and thickness, yield stress, water depth, and seabed soil stiffness. It is not a function of the number of piles. Tables 3 and 4 give values of this ratio for two yield stress levels and several pile sizes. Appendix A provides additional guidance. The force/energy ratio for commercial fender elements is in manufacturer's catalogs.

(b) Using the energy required to be absorbed by the dolphin structure from step (a), follow the steps given in Reference (1) to select the number and size of piles required for the dolphin.

(c) For the number and size of piles determined in step (b), follow the steps given in Reference (1) to determine the dolphin force corresponding to the dolphin rated energy.

(e) Using the force determined in step (c), determine the dolphin slope change which must be accommodated.

(e) The fendering energy is simply the total energy minus that absorbed by the dolphin structure. Using this energy, the force found in step (c), the slope change found in step (d), information on approach angles, tidal range, and hull form, design the fendering.

(f) Iterate, if necessary, to correct for any errors in initial assumptions.

11. RECOMMENDATIONS FOR FENDER DEVELOPMENT EFFORT

11.1 Development of Tire Stack Fenders

The concept of tire stack fenders incorporating (used) large diameter tire casings, which has been so successfully applied by the Toronto Harbour Commission, appears to be a very economical fender system. It is widely applicable for dolphin sub-assemblies, applicable as fendering for piers, and appears to be an economical application of material which is in oversupply. As steps toward reliable implementation of this concept it is recommended:

a) A survey of the tire industry for the purpose of cataloging tire casing sizes, characteristics and relative availabilities.

b) Tests of several representative sizes of large diameter air-inflated tire casings to establish load-deflection relationships and load/contact-area relationships as functions of tire geometry and inflation pressure. This will supplement the rather limited data available in Firestone Burleigh catalogs.

c) Tests similar to (b) for tire casings packed with shredded rubber and, possibly, for foam-filled casings.

d) Preparation of tables, based on (b) and (c) to facilitate the fender selection process.

11.2 Development of New Concepts Employing Tires as Energy-Absorbing Fender Units

It is very clear that there are other, potentially useful, ways to use tire casings in fenders. These include, for example, submerged tires which, under loading, expel water through orifices of proper size to produce resisting force which is a function of the rate of deflection. Apart from the fact that such elements would be rebound-free, they would have advantageous energy-absorbing characteristics, particularly when used on flexible dolphins, but in other fendering conditions as well. These and other applications of tire casings to fendering should be tested. Reference (10), for example, describes one application which merits serious development effort.

11.3 Development of Pile-Supported, Floating Mooring Platforms

The force/displacement and energy-absorbing characteristics of steel cylinder piles now are well understood, and such piles have been widely employed for dolphins throughout the world. However, there are opportunities for applying this knowledge to the design of floating mooring platforms which have, as yet, been much less developed than dolphins. Reference (9) describes one such platform which merits serious study.

REFERENCES

(1) Hansen, Holley and Biggs, "Design Standards for Structural Steel Dolphins - Phase IV," NAVFAC Contract N00025-71-0023, Nov., 1973.

(2) "Marine Fenders," Volume 2, Load Kinematics, Lord Corporation, Erie, Pa.

(3) T. T. Lee, "A Design Criteria for Marine Fender Systems," Proc. of 11th International Conference on Coastal Engineering, London, Sept. 1968.

(4) NAVFAC DM-25 Waterfront Operational Facilities, dated Oct. 1971.

(5) Hansen, Holley and Biggs, "Investigation of Sleeved Piles for Steel Dolphins - Final Report," NAVFAC Contract N00025-71, July 1974.

(6) NAVFAC DM-26 Harbor and Coastal Facilities, dated July 1968.

(7) Firestone Burleigh Catalog-Marine Fendering.

(8) Schmid, W.E., Elms, D.G., Carioti, B.M., and R.C. Peace, "Analysis and Design of Dolphins," Dept. of Civil Engineering, Princeton University, Princeton, N.J., 1973.

(9) C. J. Kray, Disclosure of "Floating Articulated Docking and Mooring Platform - Camel," July 1975.

(10) C. J. Kray, Disclosure of "Economical Hydraulic Fender Pile," September 1975.

(11) NAVFAC DM-2 Structural Engineering, dated Sept. 1974.

(12) ---- "Novel Fender Withstands Seaway Rigors," American Seaport, October 1976.

APPENDIX A - EFFECTS OF PILE YIELD STRENGTH LEVEL ON DESIGN OF DOLPHINS AND DOLPHIN FENDERING

A.1 Principal Steel Dolphin Pile Characteristics

Multi-pile steel dolphins are provided with interpile connections which force the several piles to deflect equally, and to share equally the applied force and the absorbed energy. Thus the total energy absorbed by the dolphin structure (that is, the total energy minus energy absorbed by fender and fender mounting) is simply the energy absorbed by a single pile multiplied by the number of piles. Similarly, the total dolphin force associated with a given level of energy absorption is simply the force per pile multiplied by the number of piles.

In general it is desirable to use as few piles as possible in a multi-pile dolphin. The less piles that are required, the less costly are the required interpile structural connections, and the more physically compact is the dolphin. The latter consideration may be important when the space available for a dolphin is limited. Obviously the single-pile dolphin represents the ultimate in simplicity. The required number of dolphin piles may be reduced by using stronger piles (piles of larger diameter and/or thickness, piles of higher yield strength), by using a higher ratio of pile maximum stress to yield stress, by using energy-absorbing fenders and fender mountings. These approaches obviously may be used singly or in combination.

Under some circumstances the total energy to be absorbed may be so large that the required number of piles is large even when high strength piles are used, and additional energy is provided in the fendering. In other cases, the required total energy to be absorbed is more moderate, and there is a wide range of choices, of piles (number, size, yield strength), and of fender energy-absorbing elements. Under such circumstances the designer will carry through the design for two or more alternative dolphin systems and compare them on the basis of first cost and estimated maintenance costs.

From the foregoing it will be obvious that the energy-absorbing capacity of a dolphin pile is one of the characteristics of dominant importance to the designer. However, the pile force required to achieve a given level

of energy absorption also is an important characteristic. This force dictates the fender contact area which must be provided to avoid excessive pressures on the vessel hull. It likewise influences the strength of the fender and fender connection details required. It is for this reason that the force/energy ratio, F/W, is a widely mentioned characteristic in the literature on commercially available fender elements. This ratio is a measure of the effectiveness of a particular energy-absorbing fender element or, indeed, the effectiveness of a total dolphin system. By this measure, the lower this ratio, the more effective is the element or dolphin system.

Of somewhat lesser importance, but involved in the design process, are the pile-top deflection and pile-top slope which are associated with the achievement of a given level of energy absorption. For example, the minimum distance that the fender contact surface must be placed, outboard of the piles, in order to avoid vessel contact with the unprotected lower portions of the dolphin piles, depends upon the pile deflection (and upon deflections within the fender system). The relative vessel-to-dolphin angles which must be accommodated by articulations within the fender mounting are dependent (in part) on the pile-top slope.

It is of interest to note that, for a given pile energy absorption, W, the associated force, F, is inversely proportionate to the associated pile-top deflection, Δ. Since the pile deflection is assumed to be linearly proportionate to the force, F, (a conservative simplification), we write for a pile

$$W = \frac{1}{2} F \Delta \qquad \text{(A1)}$$

and

$$\frac{F}{W} = \frac{2}{\Delta} \qquad \text{(A2)}$$

Thus an improvement in the force/energy ratio, F/W, implies an increase in the deflection, Δ, and the pile-top slope). Nevertheless, the advantages associated with a reduction in F/W far outweigh the costs of accommodating the increased pile deflection and pile-top slope.

A.2 Dependence of Dolphin Pile Characteristics on Stress Level, Diameter, Thickness, Water Depth, and Seabed Soil Stiffness.

As discussed in the preceding Section, the characteristics of dolphin pile response of principal interest are:

(a) energy absorption
(b) force associated with energy absorption
(c) force/energy ratio
(d) pile-top deflection associated with energy absorption
(e) pile-top slope associated with energy absorption.

Based on Appendix A in Reference (1) we write the following expressions relevant to the above characteristics.

Ref. (1) Eq.		
(A8)	$M_y(\text{kip-ft}) = 3\pi f_y D^2 t_n (1 - \frac{t_n}{4D})$	(A3)
(A16)	$T(\text{feet}) = \frac{1}{12} \sqrt[5]{\frac{EI}{f}}$	(A4)
(A17)	$I = 216\pi D^3 t \, (1 - \frac{t}{4D})$	(A5)
(A20)	$M = C_M F H$	(A6)
	$\Delta = C_\Delta \frac{MH^2}{EI}$	(A7)
	$\theta = C_\theta \frac{MH}{EI}$	(A8)
(A29)	$W_R = C_E A_E H$	(A9)

$$C_E = \frac{2(.75\ f_y)^2}{30{,}000} \frac{C_\Delta}{C_M} \qquad \text{(A10)}$$

$$A_E = \frac{1.5\pi D t_n^2 \ (1 - t_n/4D)^2}{t(1-t/4D)} \qquad \text{(A11)}$$

where f_y = yield stress, ksi
D = pile outside diameter, ft.

t = pile wall nominal thickness, ins.
t_n = t - 1/8", ins.
T = pile characteristic length, ft.
E = modulus of elasticity, ksi, $k/ft.^2$
I = pile moment of inertia in^4, ft^4
f = coefficient of horizontal/subgrade reaction kips/in^3, kips/ft^3
Δ = pile-top deflection, ft
θ = pile-top slope, radians
C_Δ = coefficient, dimensionless
C_θ = coefficient, dimensionless
M = maximum pile bending moment, kip-ft.
M_y = yield bending moment
H = water depth, ft.
C_E = coefficient, kip/in^2
A_E = coefficient, in^2
C_M = coefficient, dimensionless

In the expression for absorbed energy, W_R, consider, first, the coefficient, A_E, defined by Eq. (A11), and plotted on Fig. 5 of Ref. (1) as a function of D. Examination of Eq. (A11) reveals that this coefficient increases approximately in direct proportion to the pile wall thickness and pile diameter. Next, consider the coefficient C_E, which also appears in the expression for absorbed energy. This coefficient is defined by Eq. (A10), and is plotted on Fig. 4 of Ref. (1) as a function of T/H. It varies directly with the square of the maximum bending stress, which is taken as 0.75 f_y. Thus if f_y is doubled, the absorbed energy is multiplied by $(2.0/1.00)^2$; i.e., by 4.00. In summary, the choice of steel strength grade and the proportion of yield strength utilized are the dominant factors influencing absorbed energy.

From Eq. (A6) we write

$$F = \frac{M}{C_M H} \tag{A12}$$

The coefficient, C_M, plotted on Fig. 6 of Ref. (1) against T/H, is a function of the relative stiffnesses of pile and soil. However, this coefficient is very insensitive. Thus, the force F is almost directly proportionate to the maximum pile bending moment. This in turn increases directly with bending stress (see Eq. (A3)), directly with the square of pile diameter, and directly with pile wall thickness.

Since the force and absorbed energy increase directly with the stress and the stress squared, respectively, it follows that the force/energy ratio, F/W, increases in inverse proportion to the stress. Thus a doubling of the steel yield strength implies a 50 percent reduction in F/W. Similarly if it is decided to use 1.0 f_y (rather than 0.75 f_y) under infrequent overload, this implies a 25 percent reduction in F/W. Since the development of a maximum stress equal to the yield stress does not imply impending failure (there is reserve strength in the range of plastic behavior), this option may be attractive in adverse circumstances.

The coefficients C_Δ and C_θ , in Eqs. (A7) and (A8), which are plotted against T/H in Fig. 7 of Ref. (1), are dependent on the relative stiffnesses of pile and soil. They are relatively insensitive to changes in soil stiffness, because T varies inversely to the one-fifth power of the soil stiffness (see Eq. (A4)). They are only moderately sensitive to changes in pile stiffness (i.e., changes in pile diameter and thickness). Thus pile-top deflection and slope, as given by Eqs. (A7) and (A8), are primarily dependent on, and directly proportionate to (M/EI). Eqs. (A3) and (A5) show that M varies directly with pile diameter squared and almost directly with thickness, while I varies directly with diameter cubed and almost directly with thickness. Therefore (M/EI) varies inversely with diameter. It is obvious from Eq. (A3) that (M/EI) also varies directly with stress. In summary, if we increase the yield strength of the steel, or the portion of yield strength utilized, we increase pile-top slope and deflection proportionately. If we increase the pile diameter we decrease pile-top slope and deflection. Normally the advantages of increased stress more than offset the problems associated with increased slope and deflection.

Throughout the foregoing discussion the dominant influence of the steel yield stress has been obvious. Thus, if one wishes to increase the pile

capacity (in order to use fewer piles), the piles can be made larger, thicker, or of higher strength steel. The latter approach usually is the most effective and, in addition, this approach lowers the force/energy ratio. There will, of course, be occasions when high strength steel piles are not economically available, and increasing pile capacity will require increasing diameter and/or thickness.

A.3 Illustrative Examples

Example 1

A dolphin is to be designed to absorb a total energy of 380 kip-ft. under loading applied 60 feet above the seabed. It is assumed that 80 kip-ft. of energy absorption will be provided by the fendering. Determine the number of piles required if f_y = 60 ksi, maximum stress is 0.75 f_y, and pile diameter and thickness are 4.0 ft. and 1.0 inches. Assuming the capacity of the piles is fully developed, determine force, F, the W and F/W values to be satisfied by the fendering, and the pile-top slope and deflection.

From Ref. (1), Fig. 3, for D = 4.0 ft. and t = 1.0 ins.,

$$T_{min} = 11.4 \text{ feet}$$

$$\frac{T_{min}}{H} = \frac{11.4}{60} = 0.190.$$

From Ref. (1), Fig. 4, for T_{min}/H = 0.190 and f_y = 60 ksi,

$$C_E = 0.095 \text{ ksi.}$$

From Ref. (1), Fig. 5, for D = 4.0 feet and t = 1.0 inch,

$$A_E = 13.6$$

$\therefore$ The energy-absorbing capacity of N piles is:

$W = N(.095)(13.6)(60) = 77.5\ N$

$N = \frac{380-80}{77.5} = \underline{N = 3.9 \text{ piles}}$

$\therefore$ <u>4 piles required</u>

Full energy capacity of piles = W = 77.5(4) = <u>310</u> kip-ft.

Net energy capacity required of fendering

= 380 - 310 = <u>70 kip-ft</u>.

From Eq. (A3)

$M_y = 3\pi(60)(4)^2(0.875)(1 - \frac{0.875}{4(4)}) = 7480$ kip-ft.

Since allowable stress = 0.75 f_y, maximum pile moment = 0.75(7480) = 5610 kip-ft.

From Fig. 6 of Ref. (1), $C_M = 1.06$

From Eq. (A6)

$4610 = 1.06\ F(60)$

$\therefore$ F, per pile $= \frac{5610}{1.06(60)} = 89$ kips/pile

For 4-pile dolphin, <u>F = 4(89)= 356 kips</u>

Reqd. $(F_W)_{fendering} \leq \frac{356}{70} = \underline{5.1}$

$(F_W)_{dolphin} = \frac{356}{380} = \underline{0.94}$

Note: The F/W of 5.1 for the fendering would be easily achieved.

From Fig. 7 of Ref. (1)

$$C_\Delta = 0.75$$

$$C_\theta = 0.84$$

From Eq. (A5)

$$I = 216\pi(4)^3(1.0)(1 - \frac{1.0}{4(4)}) = 40{,}700 \text{ inch}^4$$

Conservatively, use $M = M_y$ to compute pile-top deflection and slope. Thus

$$\Delta = \frac{0.75(7480)(60)^2(144)}{30{,}000\ (40{,}700)} = \underline{\Delta = 2.38 \text{ ft}}.$$

$$\theta = \frac{0.84(7480)(144)(60)}{30{,}000\ (40{,}700)} = \underline{\theta = 0.044 \text{ radians}}$$

Example 2

Repeat Example 1, *using* f_y *= 100 ksi.*

Since energy-absorbing capacity is proportionate to stress squared,

$$W = (\frac{100}{60})^2\ (77.5\ N) = 214\ N = 380 - 80 = 300$$

$$\therefore\quad N = \frac{300}{214} = 1.4 \qquad \therefore\quad \underline{\text{Use 2 Piles}}$$

If energy-absorbing capacity of the piles is used to absorb all the energy, what stress level is reached?

$$(\frac{f_y}{60})^2\ (77.5)2 = 380$$

$$\therefore\quad f_y = 60\ \sqrt{\frac{380}{2(77.5)}} = \underline{f_y = 94.\text{ksi}}$$

$$\text{and}\quad \max f = 0.75\ f_y \quad = \quad \underline{f = 70.5 \text{ ksi}}$$

Thus, if $f_y \geq 94.$ ksi, and $f_{max} = 0.75\ f_y$, no energy-absorbing fender elements are needed.

Since Δ and θ are proportionate to stress,

$$\Delta = (\frac{94}{60})(2.38) = \underline{\Delta = 3.73 \text{ ft.}}$$

$$\theta = (") (.044) = \underline{\theta = 0.070 \text{ radians}}$$

Since M is proportionate to stress,

$$M_y = \frac{94}{60}(7480) = M_y = 11{,}720 \text{ kip-ft.}$$

At $f_{max} = 0.75\ f_y$, M = 0.75 (11,720) = M = 8,790 kip-ft.

From Eq. (A6), $F = \frac{8790}{(1.06)(60)} = 1.38$ kip/pile

For 2-pile dolphin, F = 2(138) = $\underline{F = 276 \text{ kips}}$

Dolphin F/W = 277/380 = $\underline{0.73}$

Note that the increase in f_y has

- Eliminated the need for energy-absorbing elements in the fendering
- Reduced number of piles from 4 to 2
- Reduced dolphin force from 352 kips to 276 kips.

Example 3

For the previous examples determine if the number of piles can be further reduced by increasing f_{max} *from 0.75* f_y *to 1.0* f_y.

$$W_R = (\frac{100}{.75\text{x}60})^2(77.5)(1) = 382 \text{ kip-ft} > 380$$

$$\therefore \underline{\text{1 pile sufficient}}$$

$$M_y = (\frac{100}{60})(7480) = M = M_y = 12{,}470 \text{ kip-ft.}$$

$$F = \frac{12{,}470}{1.06(60)} = F = 196 \text{ kips}$$

Dolphin F = 1(196) = <u>196 kips</u>

Dolphin $F/W = \frac{196}{380} = \underline{0.52}$

Note that, by increasing f_y, and by fully utilizing f_y, we have

- Eliminated need for energy-absorbing fendering
- Reduced number of piles from 4 to 1
- Reduced the dolphin force from 356 kips to 196 kips.

APPENDIX B - DESIGN FOR JOINT ACTION OF DOLPHIN & PIER

B.1 Concept

There may be circumstances in which dolphins are to be used to permit mooring adjacent to a pier which has substantial, but inadequate, capacity to absorb the forces associated with the anticipated ship mooring operations. Under such circumstances it may be worthwhile to consider the possibility of designing the dolphins to resist a portion of the mooring forces, utilizing the pier strength to augment the dolphin capacity. Such a design will require a connecting strut between the dolphin and the deck of the pier. If such a strut does not interfere with pier-to-ship operations, it may lead to a sufficient reduction in the dolphin cost to merit consideration. However, there are limits to what can be accomplished by such a design. The factors discussed below are relevant.

B.2 Strength and Stiffness of Pier

It will be necessary to assess the magnitude of the force which can be applied to the pier without overstressing any of its structural components. It also will be necessary to assess the stiffness of the pier; i.e., the ratio of applied force to horizontal pier deflection. The amount of energy that can be absorbed by the pier structure is simply one-half the product of the maximum acceptable force and the deflection corresponding to this force. In most cases this energy will be much less than is required to be absorbed in anticipated ship mooring operations; otherwise the introduction of dolphins would not be contemplated.

The magnitude of the pier deflection corresponding to the maximum acceptable pier force is of particular importance. In most instances this deflection will be much less than the dolphin deflection when the design capacity of the dolphin is fully developed. This follows from the fact that steel dolphin piles are deliberately designed to act as flexible vertical beams cantilevered from the seabed with their deflection maximized to provide maximum energy absorption. In contrast most pier structures develop their allowable stresses at deflections which are small, sometimes as little as one-tenth the magnitude of the deflection of a typical dolphin pile.

B.3 Magnitude of Energy Absorption Increase in Dolphin Piles Compared with Energy Absorbed in Strut and/or Pier Structure

The vertical distance from the strut to the point of ship impact on the dolphin usually will be a relatively small fraction of the effective length of the dolphin piles. That is, the strut force acts upon the dolphin at almost the same elevation as does the ship-to-dolphin force. This fact has two important implications. First, the deflection of the dolphin, at its point of connection to the strut, is not much less than the deflection of the dolphin at the elevation where the ship impacts. Second, the energy absorbed by the dolphin piles is essentially dependent upon the dolphin deflection at the point of application of the ship impact, independent of the presence of a counteracting force from the pier-to-dolphin strut. Thus any additional energy-absorbing capacity represents, mainly, either strain energy in the pier structure or strain energy in a flexible strut, if such a strut is used. That is, there is very little additional strain energy in the dolphin piles themselves as a consequence of the presence of the strut force.

To illustrate the above, consider the two simplified cases illustrated in Fig. 15. Fig. 15(a) shows a pile fixed at a distance ℓ below the point of load application, P_o. The moment at the bottom, $M_o = P_o\ell$. Fig. 15(b) shows the same pile, loaded with the force 1.5 P_o and with a strut reaction R = 0.625 P_o, located at 0.2 below the top. This pile has the same base moment, M_o. For the pile without a strut the deflection at the top is

$$\Delta_{10} = \frac{P_o\ell^3}{3EI}$$

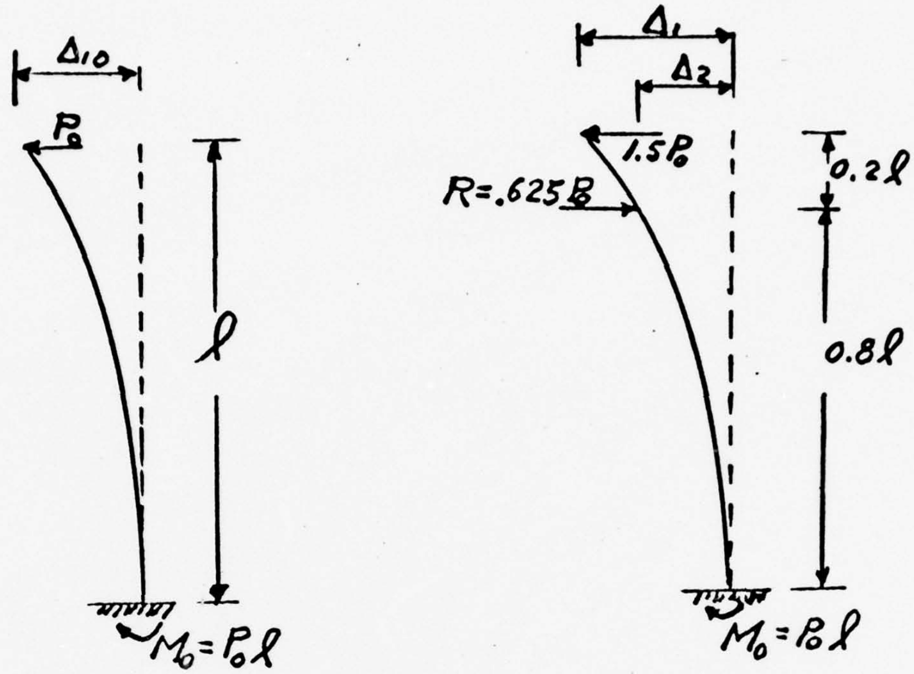
and the absorbed energy is

$$W_o = \frac{1}{2} P_o \Delta_{10} = \frac{P_o^2\ell^3}{6EI}$$

For the pile with a strut, the deflection at the top, and the total absorbed energy, are

$$\Delta_1 = \frac{1.06\ P_o\ell^3}{3EI} = 1.06\ \Delta_{10}$$

$$W = \frac{1.5\ P_o}{2}\left(\frac{1.06\ell^3}{3EI}\right) = \frac{1.59\ P_o^2\ell^3}{6EI} = 1.59\ W_o$$



(a) (b)

$$\Delta_{10} = \frac{P_0 \ell^3}{3EI}$$

$$W_0 = \frac{P_0}{2}\Delta_{10} = \frac{P_0^2 \ell^3}{6EI}$$

$$\Delta_1 = \frac{1.06 P_0 \ell^3}{3EI} = 1.06\,\Delta_{10}$$

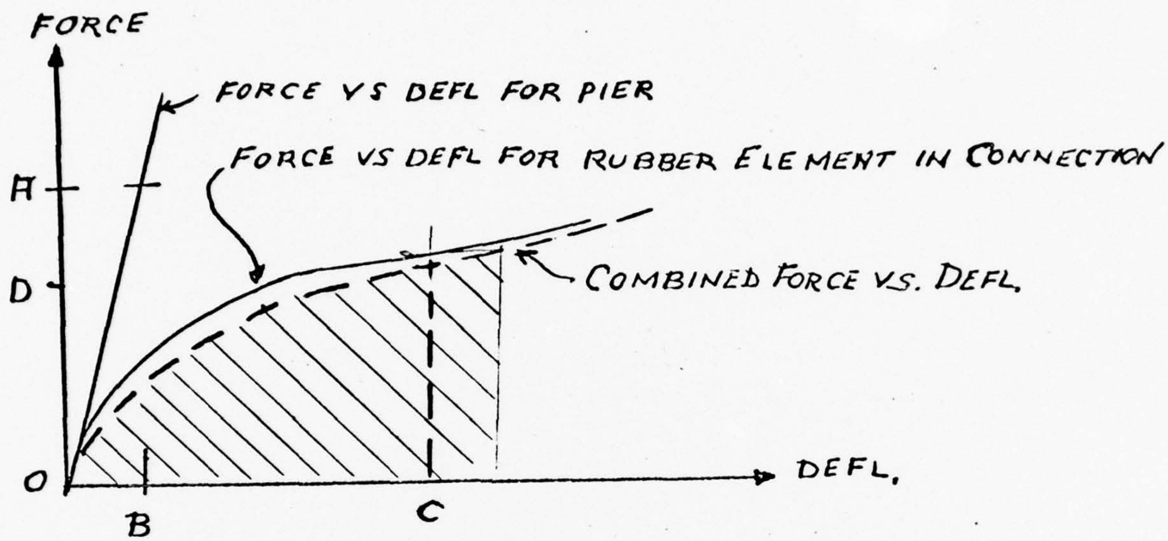
$$W = \frac{1.5P_0}{2}\Delta_1 = 1.59\,W_0$$

$$\Delta_2 = \frac{.736\,P_0 \ell^3}{3EI} = .694\,\Delta_1 = .736\,\Delta_{10}$$

$$W_S = \frac{R}{2}\Delta_2 = \frac{0.46 P_0^2 \ell^3}{6EI}$$

$$W - W_S = 1.13\,W_0$$

FIG 15 – EFFECT OF PIER-TO-STRUT FORCE ON DOLPHIN ENERGY ABSORPTION



OA = ALLOWABLE FORCE ON PIER
OB = " DEFL. OF PIER
OC = DOLPHIN DEFL.
OD = FORCE (ON ELEMENT AND PIER)
CORRESPONDING TO DOLPHIN DEFL.

CROSS-HATCHED AREA IS THE COMBINED ENERGY ABSORBED BY PIER STRUCTURE AND BY CONNECTING FLEXIBLE ELEMENT

FIG 16 – COMBINED FORCE VS DEFLECTION CURVE OF PIER STRUCTURE AND CONNECTING FLEXIBLE ELEMENT

FIG 16

i.e., a 59 percent increase in total absorbed energy. However, for this case the deflection at the strut is

$$\Delta_2 = 0.736 \frac{P_o \ell^3}{3EI} = 0.694 \Delta_1$$

Thus the strut (or the pier structure) absorbs an amount of energy

$$W_S = \frac{1}{2} R_S \Delta_2 = \frac{0.625 \, P_o}{2} \left(\frac{0.736 P_o \ell^3}{3EI}\right)$$

$$= 0.46 \left(\frac{P_o^2 \ell^3}{6EI}\right) = 0.46 \, W_o$$

Thus the energy absorbed in the pile is 1.59 W_o - 0.46 W_o = 1.13 W_o. That is, the strut has increased the pile strain energy by only 13 percent. The remaining 46 percent increase represents strain energy in the strut or in the pier structure. Finally, it should be noted that the vertical distance between applied load and strut typically would be less than 0.2 times the pile effective length. As this distance is decreased the augmentation of energy in the pile is further decreased. For example, if a = 0.1 , and a load of 1.5 P_o is again applied, the strut force R = 0.556 P_o to maintain the base moment equal to $P_o \ell$. In this case the total energy is increased 54 percent, comprised of 48 percent in the strut or pier structure and only 6 percent increase in the pile energy.

B.4 Alternative Dolphin-to-Strut Characteristics

One can consider three different types of struts, with three different force-displacement characteristics. These are:

(a) Rigid strut, positively-connected to pier and dolphin (forcing essentially equal deflections at the pier and at the strut-to-dolphin connection).

(b) Rigid strut positively-connected at one end and slot-connected at the other end (permitting a prescribed unresisted deflection of the dolphin before pier resistance is mobilized; equal increments of deflection thereafter).

(c) A strut and flexible element (e.g., rubber element) (permitting the dolphin to deflect more than the pier).

Strut type (a) would be appropriate only if the pier is very flexible. That is, it can tolerate the typical large dolphin deflections without overstressing the pier structure. This is rarely the case.

Strut type (b) would be appropriate only if the pier can tolerate a deflection which is a significant fraction, say 30 percent or more, of the deflection of the dolphin at the dolphin rated loading. In addition, of course, the energy-absorbing capacity of the pier structure should be a substantial fraction of the total energy required by the ships serving the facility. It should be noted that it may not be possible to predict the stiffness of the pier and the dolphin (which depend upon uncertain seabed soil characteristics) with great accuracy. For this reason it may be necessary to adjust the size of the "gap" (e.g. slotted connection) after the dolphin is installed, and to base such adjustment on a test. For example, deflections of pier and dolphin could be observed during a load by jacks between the two. This is likely to be a demanding, and costly, exercise.

If the pier structure can tolerate a deflection which is a significant fraction of the dolphin deflection, say 30 percent or more, and if the energy-absorbing capacity of the pier structure is a substantial fraction of the required total energy, strut type (c) may be applicable. In this case energy absorption can be gained in the pier structure and in the flexible element incorporated in the strut. It may be practical to use, as the strut flexible element, one of the commercially available rubber fender elements which rises rapidly to a peak force and then buckles, maintaining this force essentially unchanged with increasing deflection. The selected rubber element should have a peak force somewhat less than the acceptable pier force.

Strut type (c) also may be applicable when the pier can accept a very large force, but can tolerate only a very small deflection; i.e., the pier is very stiff, as well as strong, and thus cannot absorb substantial energy. In this case the additional energy will be developed primarily in the strut flexible element.

B.5 Flexible Elements in Pier-to-Dolphin Strut vs. Flexible Elements in Dolphin Fender

If most of the additional energy-absorbing capacity is developed by the flexible elements of the strut, rather than in the pier structure, it may be more reasonable not to use a strut but rather to incorporate the same (or equivalent) flexible element in the dolphin fender. If the element is located in the fender, the maximum ship-to-dolphin force will be the dolphin shear force associated with its rated energy conditions. If the flexible element is located in a strut, the maximum ship-to-dolphin force will be the dolphin shear force plus the strut force, and a larger fender shield will be required.

Under most conditions the foregoing consideration will weigh heavily against the use of a dolphin-to-pier strut incorporating a flexible element as a principal energy-absorbing component of the system.

B.6 Recommendations and Procedure

The use of pier-to-dolphin struts, to reduce the required dolphin capacity, merits consideration only if the strut will not interfere with ship access to the pier, or with ship-to-pier operation, and only

a) when the strength of the pier structure and the acceptable pier deflection imply a pier energy-absorbing capacity which is a substantial fraction of the total required energy absorption, and/or

b) when the distance from pier to dolphin (e.g., as governed by seabed conditions) will accommodate an energy-absorbing flexible element, and this location of the element (rather than in the fender) will permit mooring the ship closer to the pier, and,

c) when the cost of providing for the larger ship-to-dolphin force, occasioned by mobilizing the energy-absorbing capacity of pier structure and/or flexible interconnection (rather than providing additional capacity in flexible fender elements) does not outweigh other considerations.

The procedure to be followed in assessing whether pier-to-dolphin interconnection may be advantageous involves the following steps:

a) Evaluation of the maximum acceptable pier deflections, as limited by stresses in the structure, by the effect of deflections on the integrity of utility lines and equipment, and by the effect of deflections on the safe conduct of pier operations.

b) Evaluation of the approximate dolphin deflection, which is primarily a function of water depth, dolphin pile diameter, grade of steel, and seabed soil stiffness. Note, however, that the deflection is not sensitive to either the number of piles or the pile wall thickness. Thus, it is sufficient to determine the deflection of a single pile, of the proposed diameter and any thickness, loaded to produce a maximum bending stress of 0.75 times the yield stress.

c) Selection of a (rubber) flexible element to interpose between pier and dolphin. Depending upon the spacing of the pier and dolphin, either the dolphin may bear directly on this element or a strut may be required. This element should be of the buckling type (e.g., axially-loaded hollow cylinder; arch) which will r[illegible]n-tially constant force through large deflections. The buckling load should be somewhat less than the force required to develop the acceptable pier deflection. The element deflection capability must be in excess of the difference between the dolphin deflection and pier allowable deflection. Figs. 2-12 and 2-14 of NAVFAC DM-25 provide characteristics of rubber elements of suitable types, and there is additional similar data available from manufacturers of rubber fender elements.

d) Determination of the energy absorbed by pier structure and by the pier-to-dolphin flexible connection at the dolphin deflection computed in (b). This requires a plot of the sum of pier deflection and element deflection against force, and reading from this plot the value of force corresponding to a deflection sum equal to the dolphin deflection. For the force thus determined the combined energy absorption of pier structure and flexible connecting element

is simply the area under the combined force-deflection curve. Alternatively the energy absorbed by the element can be found from Figs. 2-12 and 2-14 of NAVFAC DM-25, and the energy absorbed by the pier structure (0.5 times the product of force and pier deflection) can be separately determined. Fig. 16 illustrates the procedure.

e) Design of dolphin and dolphin fendering for the net energy: i.e., the total required energy absorption minus the energy absorbed by pier structure and pier-to-dolphin interconnecting element.

f) Design of dolphin without pier-to-dolphin connection.

g) Comparison of (e) and (f), with respect to costs and operational convenience.